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Conveying Machines

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ТРАНСПОРТИРУЮЩИЕ МАШИНЫ

Издательство «Машиностроение» • Москва

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Conveying Machines

In two volumes

Volume II

Translated from the Russian
by
V. Afanasyev

Mir Publishers • Moscow

First published 1985
Revised from the third 1983 Russian edition

The Russian Alphabet and Transliteration

А а	a	К к	k	Х х	kh
Б б	b	Л л	l	Ц ц	ts
В в	v	М м	m	Ч ч	ch
Г г	g	Н н	n	Ш ш	sh
Д д	d	О о	o	Щ щ	shch
Е е	e	П п	p	Ъ ъ	”
Ё ё	e	Р р	r	Ы ы	y
Ж ж	zh	С с	s	Ь ь	’
З з	z	Т т	t	Э э	e
И и	i	У у	u	Ю ю	yu
Й й	y	Ф ф	f	Я я	ya

The Greek Alphabet

Α α	Alpha	Ι ι	Iota	Ρ ρ	Rho
Β β	Beta	Κ κ	Kappa	Σ σ	Sigma
Γ γ	Gamma	Λ λ	Lambda	Τ τ	Tau
Δ δ	Delta	Μ μ	Mu	Υ υ	Upsilon
Ε ε	Epsilon	Ν ν	Nu	Φ φ	Phi
Ζ ζ	Zeta	Ξ ξ	Xi	Χ χ	Chi
Η η	Eta	Ο ο	Omicron	Ψ ψ	Psi
Θ θ ϑ	Theta	Π π	Pi	Ω ω	Omega

На английском языке

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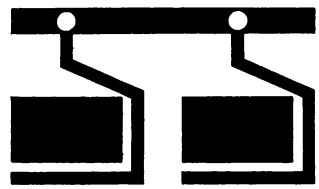
8

Overhead Conveyers



8.1. Principal Types and Applications

Overhead conveyers are used for continuous (or less frequently, periodic) transportation of unit loads (and sometimes, bulk loads) along closed paths of an intricate, mostly spatial, configuration. They are called 'overhead conveyers' because the load is carried on shelves or in boxes suspended from trolleys or carriages which move on an overhead (usually monorail) track.

Depending on the method of connection of the hauling chain with load-carrying trolley attachments and the method by which the load is moved, overhead conveyers can be divided into the following principal types:

1. *Overhead load-carrying trolley conveyers* (Fig. 8.1*a*) in which the trolleys with load-carrying attachments are fastened to the hauling chain and move on a continuous overhead track, while the chain moves along that track.

2. *Overhead pusher conveyers*, or power and free conveyers, (Fig. 8.1*b*) in which the load-carrying trolleys are not fastened permanently to the hauling chain. The chain and trolleys move on separate overhead tracks (the trolley track may have various sideways), and the chain has pushing members (pushers) which can engage the trolleys and propell them on their track.

3. *Overhead push-carrying conveyers* which are essentially the combination of a trolley and pusher conveyer. In this type of conveyers, the trolleys are fastened to the hauling chain and have hook-shaped pushers with locks which can grasp the load-carrying attachments. Thus, the load-carrying attachments are moved by pushing in some portions of the conveyer path (as in a pusher conveyer) and in suspended state (on hooks of the trolleys), i.e. as in a load-carrying trolley conveyer, in other portions.

4. *Overhead load-towing (floor-type) truck conveyers* (Fig. 8.1*c*) in which the load is carried in trucks (cars) on the floor of a shop or store. A truck has a vertical mast to engage with a clamp or pusher of overhead trolleys which are fastened to the hauling chain and move on an overhead track.

5. *Overhead tow-carrying truck conveyers* which differ from the previous type in that on-floor trucks are hinged to trolleys which are drawn by the hauling chain along an overhead track. In some portions of the conveyer path, the load-carrying trucks move on the floor (as in the previous type), and in others they are lifted and transported in a suspended state (for instance, to another floor).

Notwithstanding substantial differences in methods of load conveying and methods of connection of the hauling chain with trolley attachments, many elements of all types of overhead

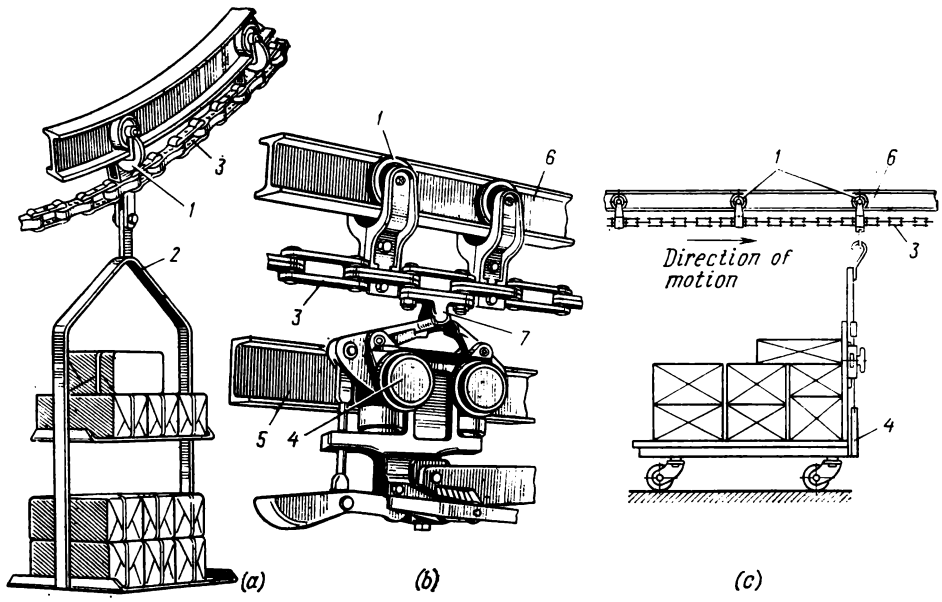


Fig. 8.1. Schematics of overhead conveyers
(a) load-carrying trolley conveyer; (b) pusher trolley conveyer; (c) load-towing truck conveyer; *1*—trolley; *2*—load-carrying attachment; *3*—chain; *4*—trolley or truck; *5*—carrying track; *6*—hauling track; *7*—pusher

conveyers have much in common. An important problem is to standardize them as much as possible. All these types of conveyers have the same design of hauling chains, drives, and turning and take-up devices, and only tracks, trolleys, and trucks have certain differences.

Overhead conveyers may be of single- or multiple-drive type and have chains (in most cases), or steel cables as the tractive elements. Overhead conveyers can carry miscellaneous unit loads of various shape and dimensions (with a length from a few millimetres to 12 m) and mass (from fractions of a kilogram to 8 tons), such as workpieces, semi-finished articles, machine parts and assembly units of machines, instruments, radio and television sets, wheels, tyres, boxes, bales, coils, etc.

Trolley attachments can be loaded and unloaded on a moving conveyer manually or automatically. The closed path of an overhead track of a conveyer allows load transportation almost along its entire length, with few unloaded path sections. The load units suspended on a conveyer can pass when being conveyed through various process operations: mechanical cleaning in sand-blasting or shot-blasting chambers, washing or etching in etching baths, painting and drying, heat treatment, storage, sorting, assembling, etc. The speed of trolley conveyers ranges from 0.1 m/min to 30 m/min, and that of the truck type may reach 45 m/min.

Overhead conveyers compare favourably with other types of continuous conveying machines in the following features: (a) their path may be of any intricate configuration in space (the moving portion of overhead conveyers can move loads in any direction) and of substantial length (up to 500 m in single-drive conveyers and up to 3 km in multiple-drive types), so that a single conveyer can carry goods through a complete production cycle with the process operations being performed on various floors of a building or in a number of adjacent buildings;

(b) the path of a conveyer can be eas-

ily adapted to possible variations in production process;

(c) an overhead conveyer can form a moving reserve of goods, which makes it possible to eliminate intermediate stores in a shop and thus save much room and labour in material-handling operations;

(d) overhead conveyers are characterized by a low power consumption;

(e) they are readily suitable for automatic control and automatic loading, unloading, and distribution of loads.

Owing to these advantages, overhead conveyers are the main and most popular type of conveyer for transportation of miscellaneous goods in and between shops, and may be used as flow-line conveyers in machine-building, metal-making, radio engineering, textile, light, rubber industries, etc.

8.2. Overhead Load-carrying Trolley Conveyers

8.2.1. General Design

An overhead load-carrying trolley conveyer (Fig. 8.2) has an endless hauling chain *3* closed over the conveyer contour, with trolleys *7* fastened to the chain and load-carrying attachments *6* hinged to the trolleys. Load units *9* to be conveyed are placed or hooked onto the attachments. The trolleys are drawn by the chain along a closed overhead track *8* which is fastened either to building or conveyer structures.

The hauling chain is moved by an electric motor drive *5*. The chain is flexible in the horizontal and vertical plane and can move along a path of spatial configuration, i.e. the overhead track can have vertical and horizontal bends in any direction. In the horizontal plane, the chain is bent on turning devices *4* and in the vertical, follows the bends *2* of the track *8*. The initial tension of the chain is formed by a take-up *1*.

The principal parameters of overhead trolley conveyers based on detachable chains with a pitch of 80, 100 or 160 mm are standardized as shown in Table 8.1.

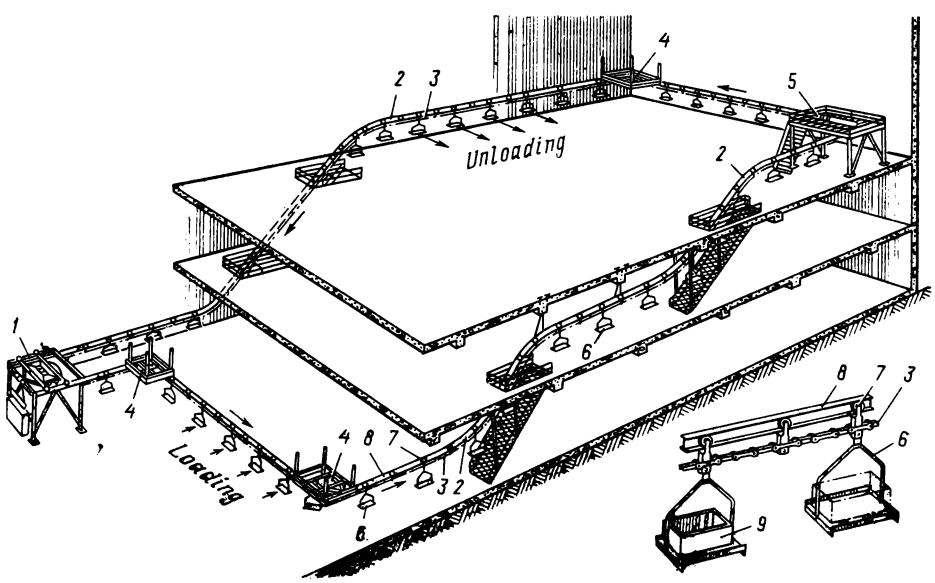


Fig. 8.2. Overhead load-carrying trolley conveyer

Table 8.1. Characteristics of Overhead Load-carrying Trolley Conveyers Manufactured in the USSR

Conveyer type	Maximum trolley load, kN	Trolley wheel diameter, mm	Hauling chain			Track	
			type	pitch, mm	breaking force, kN, at least	profile	height, mm
ГН-12	0.2	31	double-hinged	200	25	channel	40
ГН-200 Д-50	0.5	50	same	200	50	same	60
ЦПК-80Р	2.5	62	detachable	80	106	I-beam No. 10	100
ЦПК-100Р	5	83	same	100	220	I-beam No. 14	140
ЦПК-160Р	8	125	same	160	400	I-beam No. 16	160

8.2.2. Conveyer Elements

Tractive element. Various types of chains are mostly used for the purpose and much less frequently, steel cables. Trolley conveyers with the path lying in the horizontal plane can employ all types of chain, for instance, hauling laminated roller chains with a pitch of 65-200 mm or driving laminated chains with a pitch of 25-38 mm. Spatial-path conveyers employ chains possessing flexibility in two directions. In light-type conveyers most popular are double-hinged chains with alternate or combined axes of vertical and horizontal joints (Figs. 3.14 and 3.15), with a pitch of 160 mm or 200 mm, while in medium- and heavy-duty conveyers, use is made of hot-forged detachable chains with a pitch of 80, 100 or 160 mm (see Fig. 3.11). Welded heat-treated round-link chains with a rod diameter of 9, 11, 13, 14 or 16 mm are sometimes employed in light overhead conveyers. Sectionalized chains (see Figs. 3.13 and 3.16) have only a limited application.

With the application of double-hinged or sectionalized chains and hinged load-carrying trolley attachments in the form of a rigid triangle (or a cantilever bar), overhead conveyers can have a path with vertical ascending (inclined), and descending (declined) sections (Fig. 8.3*a* and *b*).

In other countries, light-type overhead trolley conveyers are known to employ glass fibre chains with spherical joints (see Fig. 3.16*c*).

Trolleys. A trolley of a trolley conveyer (Fig. 8.4) has two wheels *2* with cantilever axles and bearings, brackets *3* (on which the wheel axles are mounted), and washers *4*. The brackets are connected together by bolts *1* upon being passed through an internal link *5* of detachable chain. Lugs *8* on the brackets fix the position of the chain link. A bolt *6* and tube spacer *7* serve for hinged fastening of a load-carrying attachment.

A trolley conveyer may have working (single and traverse) trolleys and auxiliary support trolleys. A single working trolley is connected directly with a load-carrying attachment. Traverse trolleys are used when it is needed to transport heavy or bulky long pieces of load; then a traverse is fastened to two or four trolleys and the load is attached to it. Auxiliary trolleys are placed between the working ones on the track to prevent sagging of the hauling chain. The chain may have rigid (Fig. 8.4) or hinged (see Fig. 3.13) fastening to trolleys, the latter method being emplo-

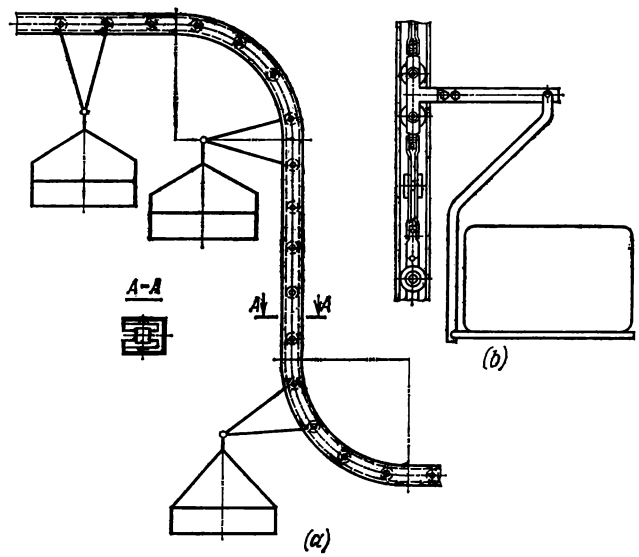


Fig. 8.3. Vertical path sections of overhead load-carrying trolley conveyer

yed to allow the trolleys to pass vertical bends of a small radius.

Wheels of trolleys are of the flangeless type. The profile of wheel rim is chosen according to the profile of the track. Wheels to run on the flanges of an I-beam are made conical, cylindrical-conical, spherical or spherical-conical (with the horizontal axles), and cylindrical or double-cylindrical (with inclined axles). Conical-rim wheels ensure steady central motion of the trolley on an I-beam, but involve additional losses due to sliding friction which appears owing to different circumferential speeds at internal and external edges of the rim in contact with the beam flange. For that reason, they are not used in modern overhead trolley conveyers. A spherical rim, with the centre of the sphere on the axis of symmetry of the wheel, cannot ensure steady central position of a trolley, so that the latter can oscillate sideways and is subject to wear in the contact zone.

In modern designs of trolleys, bearing wheels with spherical conical rims are employed; they are manufactured at ball and roller bearing plants as integral sets comprising a bearing and wheel rim. A bearing wheel (Fig. 8.5*a*) has an outer ring (wheel rim) *3*, inner ring *1*, bearing balls *4* of an increased diameter, separator *5*, cover *6*, and stamped labyrinth seal *2*. A large diameter of bearing balls (10-16 mm) allows high loads on the wheel; the gaps between bearing balls and races are increased, so that the bearing can operate reliably even on contamination of the lubricant and at ambient temperatures up to 200°C.

The type and size of bearings are selected according to the rated load and the temperature of the surroundings. As a rule, trolley wheels have rolling bearings for ease of running: standard ball bearings for temperatures up to 120°C, and roller or tapered roller bearings for temperatures up to 350°C (lubricated by dry flaky graphite). Trolley wheels of light-type overhead conveyers are sometimes made of plastics with bearings of low-friction materials. Trolley wheels are forged of steels 40, 45 or 50, with the rims heat-treated to 30-40 HRC, or cast of iron (in metal moulds). Bearing wheels are made of steel ШХ15. Plastic (polycaprolactam, etc.) wheels are known to be used in light-type conveyers.

Trolley wheels have, as a rule, labyrinth seals which reliably protect the

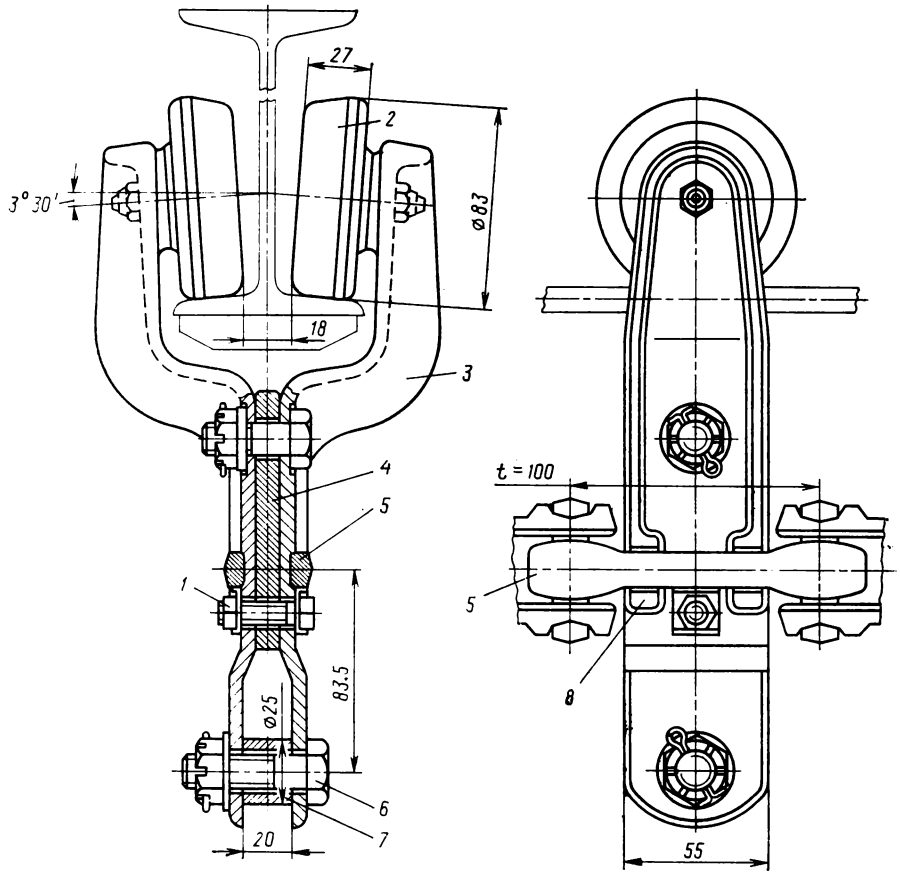


Fig. 8.4. A trolley of overhead conveyer

grease against contamination. Their bearings are packed with grease upon repair disassembling, or lubricated periodically by means of grease cups; automatic lubrication is also employed.

The brackets of trolleys should possess a sufficiently high strength and a high rigidity and have a low mass where possible. Insufficiently rigid brackets may bend and disturb proper contact between the wheels and the guide track. Brackets are forged of steel 40 or cast of steel 35JI.

The pressure on a trolley depends on the profile of conveyer path. In conveyers with the path lying in a horizontal plane without vertical bends, the pressure on a trolley carrying an attachment with the load, P_l, is the sum of the gravity forces of the heaviest piece of load, the bracket, and the chain length between the adjacent trolleys:

$$P_l = g\,(m_{l\max} + m_a) + q_{ct}t_c \quad (8.1)$$

where $g = 9.81$ m/s^2 is the acceleration due to gravity; $m_{l\max}$ and m_a are the masses of the heaviest load and the load-carrying attachment, kg; q_{ct} is the linear gravity force of the chain and trolley, N/m; and t_c is the trolley pitch, m.

The pressure P_l, of a constant magnitude and direction, is the main parameter for calculating trolley conveyers with a horizontal path.

In conveyers with a spatial path having vertical bends, the trolley in the

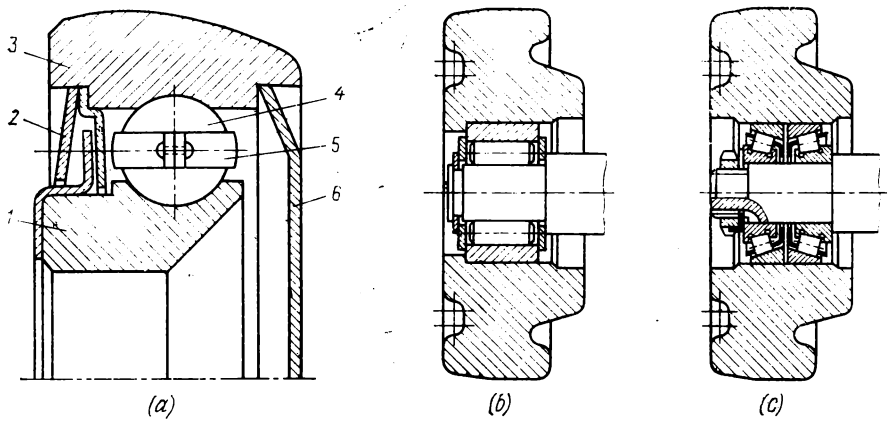


Fig. 8.5. Trolley wheels
(*a*) standard bearing-wheel; (*b*) and (*c*) special wheels with needle and tapered roller bearings (for high-temperature application)

horizontal sections is acted upon by the same force as given above by formula (8.1) and in the vertical sections, additionally by the chain tension force directed along the radius of the arc of bend. In the arc l_A (Fig. 8.6), which is convex downward, it is directed upwards and in the arc l_B, which is convex upward, the force is directed downwards.

The additional force in the arc l_A is

$$P_A = -2S_A \sin(\alpha_t/2)$$

where S_A is the chain tension at the end of arc l_A, N, and α_t is the turn angle of trolleys along a length of one pitch, deg.

Since $\sin(\alpha_t/2) \approx t_c/(2R)$, where t_c is the pitch of trolleys and R is the bending radius, we have

$$P_A = -S_A t_c/R_A = -K_t S_A$$

where $K_t = t_c/R_A$ is the coefficient of additional pressure on a trolley (Fig. 8.7).

The additional force in the arc l_B is

$$P_B = S_B t_c/R_B = K'_t S_B$$

The total force acting on a trolley in a curved section of the path will be the geometrical sum of the forces P_l and P_A or P_B.

A spatial conveyer path usually has a number of vertical bends where the chain tension may be different. Therefore, the additional and total force will also be different. They increase with an increase of chain tension and decrease of the radius of vertical bends (Fig. 8.8).

The highest force acts on a trolley in the outlet arc of a vertical bend of radius R where the chain has the highest tension S_{max}:

$$P_{max} = P_l \cos\alpha_t + S_{max} t_{max}/R \qquad (8.2)$$

This force is used to determine the strength and rigidity of axles, wheels, and trolley brackets.

In a similar manner, the force acting on an auxiliary trolley can be found.

Rolling bearings of trolley wheels are calculated for the equivalent force P_{eq} acting on a trolley, which is determined by a diagram of forces P acting on a trolley in each cycle of conveyer operation (Fig. 8.8).

For conveyers with the path lying in a plane

$$P_{eq} = \sqrt[3]{(P_l^3 L_l + P_e^3 L_e)/L} \qquad (8.3)$$

where L, L_l, and L_e are the horizontal projections of the entire path (L), the loaded section (L_l), and the empty section (L_e), and $P_e = gm_a + q_{ct} t_c$ is the force acting on the trolley in a horizontal section of the empty run.

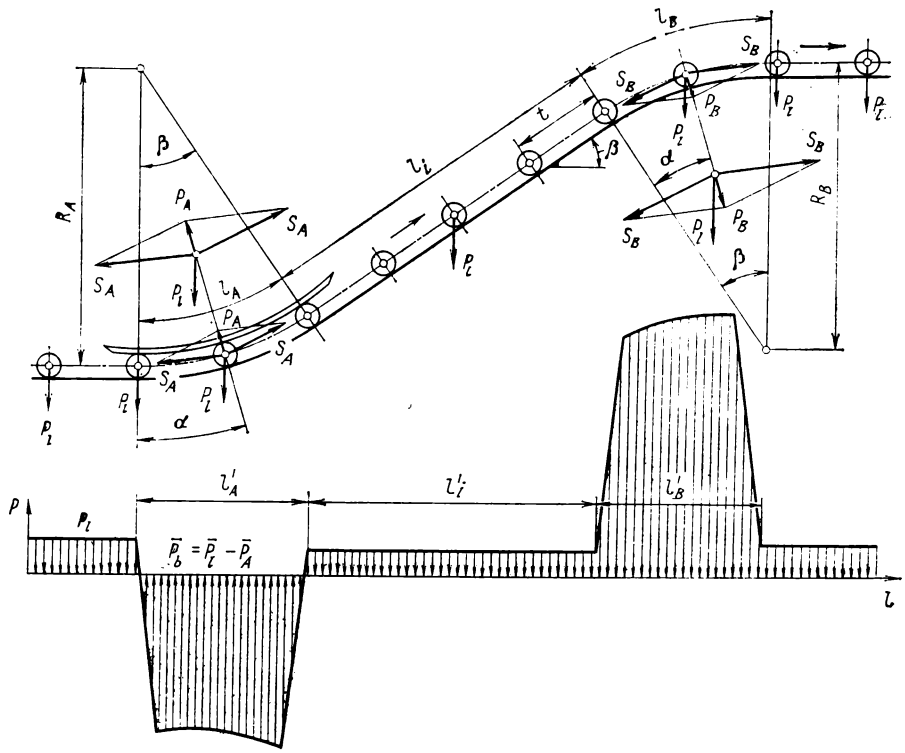


Fig. 8.6. Diagram of forces acting on trolley in vertical bend of conveyer path

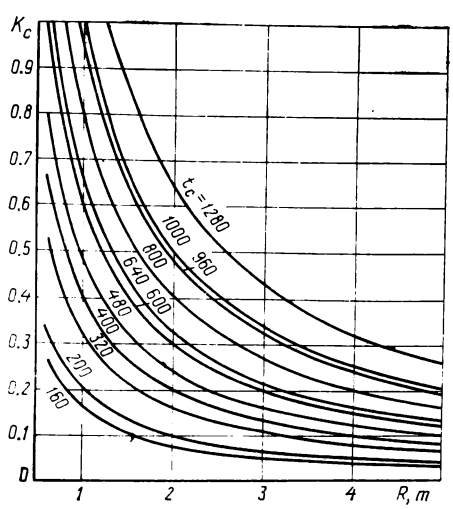


Fig. 8.7. Effect of trolley pitch and bending radius on coefficient K_c of additional forces acting on trolley

For conveyers with a spatial path

$$P_{eq} = \sqrt[3]{\frac{1}{L}\left[P_l^3 L_l' + P_e^3 L_e' + 0.0175 \sum_1^x \beta_i \left(R_A P_{Ai}^3 + R_B P_{Bi}^3 \right) + \sum P_{ii} l_{ii} \right]} \quad (8.4)$$

where L_l' and L_e' are the horizontal projections of the loaded and empty run of the conveyer path minus the horizontal projections of vertical bends in these runs; β_i is the angle of incline of the path on bending in the vertical plane; i is the number of a vertical bend in the conveyer path, with their total number being x; P_A and P_B are the forces acting on the trolley in arcs A and B of a vertical bend (see Fig. 8.6); $P_{ii} = P_l$

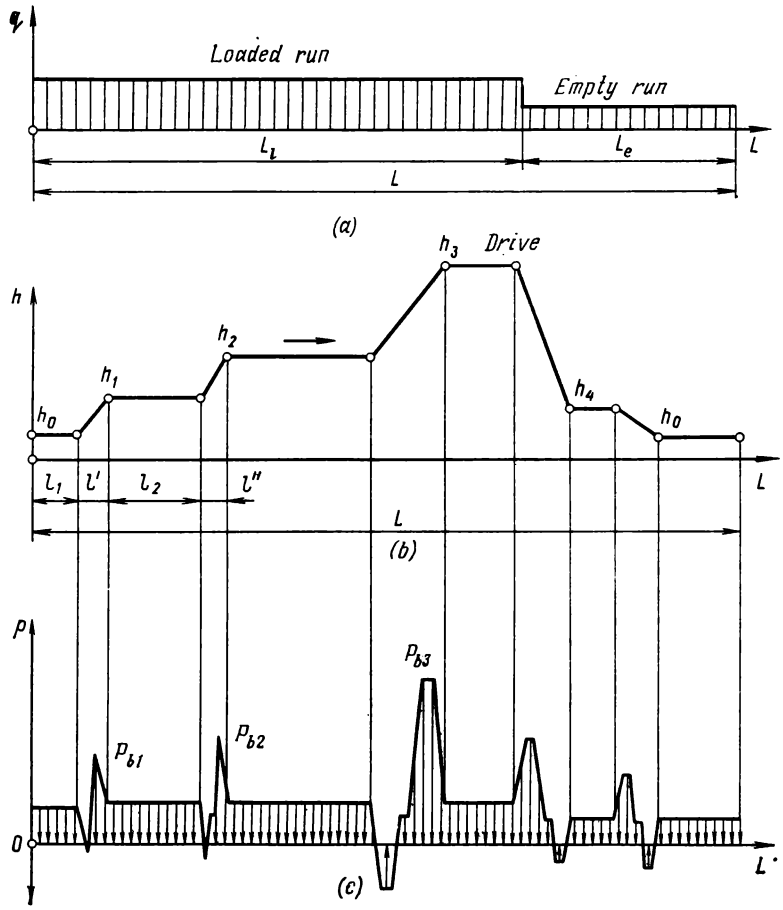


Fig. 8.8. Diagrams to calculate forces acting on trolley
(a) diagram of linear forces q along the path; (b) conveyer path profile; (c) diagram of forces acting on trolley

cos β_i is the force acting on the trolley in an i-th inclined section of a vertical bend; and l_{ii} is the length of an i-th inclined section.

The rated service life of wheel bearings is taken within 32 000-40 000 h. Wheels and bearings of double-hinged chains are calculated as trolley wheels.

Overhead trolley conveyers may have path of various configuration, different trolley speeds, and different operating conditions. Therefore, the actual forces acting on the trolleys depend on the characteristics of a particular conveyer. For the sake of unification, conveyers are designed to employ standard trolleys manufactured in series and having a rated static load P_r. This rated static load is the initial parameter for calculating the allowable useful load P_{al} for a particular conveyer, with account duly taken of its speed, and the temperature and operating conditions.

A standard trolley will be suitable for operation in a particular conveyer if the following two conditions are observed:

for the strength of trolley

$$P_r \geqslant P_{\max} \tag{8.5}$$

where $P_{\max}$ is the maximum load on the trolley determined by formula (8.1) or (8.2) depending on the path profile;

and for the load on trolley bearings

$$P_{al} \leqslant P_{eq} \qquad (8.6)$$

If these conditions are not satisfied, the load should be carried by means of a traverse on two or more trolleys or a heavier standard trolley should be employed. The magnitude of $P_{\max}$ can be diminished by increasing the radius of a vertical bend in the zone of highest tension.

The allowable load P_{al} is determined from P_r by introducing the coefficients K_1 (the ratio of the speed of the designed conveyer to the rated speed adopted for calculating the service life of bearings of a standard trolley), K_2 (to consider the temperature conditions of conveyer operation), and $K_3 = 1.1$-1.2 (the non-uniform distribution of load on trolley wheels)

$$P_{al} = P_r K_1 K_2 / K_3 \qquad (8.7)$$

The coefficients K_1 and K_2 can be found in bearing manufacturer's certificates. It is usually taken that $K_1 =$ $= 0.8$-0.6 at a trolley speed of 4-8m/min and $K_1 = 0.4$ at 30 m/min; the coefficient K_2 is equal to unity at ambient temperatures up to + 125°C and 0.7 at temperatures up to + 250°C.

The highest allowable mass $m_{l\,al}$ of the load carried by a trolley (the useful load-carrying capacity of the trolley of a given conveyer) can be found by the formulae:

for conveyers lying in a plane

$$m_{l\,al} \leqslant \frac{1}{g}(P_{al} - q_{ct} t_c) \qquad (8.8)$$

and for spatial-path conveyers

$$m'_{l\,al} = \frac{1}{g}\left(P_{al} - \frac{t_{c\,\max}}{R} S_{\max}\right) \frac{1}{\cos(\beta/2)} - q_{ct} t_{c\,\max} \qquad (8.9)$$

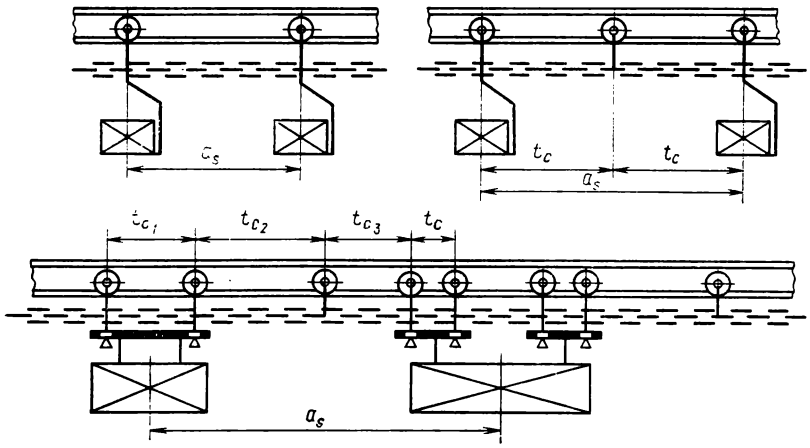
If the load to be conveyed has a higher mass than that allowed by formulae (8.8) and (8.9), it should be carried by means of a traverse on two or four trolleys (Fig. 8.9). A traverse is a shaped beam attached by ties or bolts to two or more trolleys; it has a hole in the middle for hinged attachment of a trolley or another traverse. The allowable load on a traverse can be found by the formula

$$P_{tal} = k_{tr} n P_{al} \qquad (8.10)$$

where k_{tr} is the coefficient of uneven distribution of the load on the trolleys attached to the traverse ($k_{tr} = 0.9$ for conveyers with the path lying in a plane, and $k_{tr} = 0.75$ for spatial-path conveyers); n is the number of trolleys carrying the traverse, usually $n = 2$ or 4; and P_{al} is the allowable load on a single trolley.

The pitch of trolleys (i.e. the spacing between them) is determined by the spacing between the load-carrying attachments and the required radius of vertical bends of a conveyer. With a smaller trolley pitch, a smaller radius of vertical bends can be adopted. In conveyers with vertical bends of the path, the trolleys are usually spaced at 4-10 chain pitches, but not more than 800-960 mm. With a greater spacing between the load-carrying attachments, auxiliary (supporting) trolleys should be provided between the load-carrying trolleys in order to avoid excessive sag of the hauling chain. In horizontal conveyers without vertical bends of the path, the trolley pitch can be increased up to 1200-1600 mm. Traverse trolleys are spaced in two-four chain pitches (see Fig. 8.9). The trolley pitch in chain conveyers should be a multiple of two chain pitches, since a chain may have two kinds of links, say, internal and external in detachable chains, and the trolleys can be fastened only to internal links.

Turning devices. Turns of a conveyer path in the horizontal plane (Fig. 8.10) are effected by means of turning devices, such as sprockets, sheaves, roller sets and guide rails. A particular type of turning device is chosen depending on the type of tractive element, its tension, and the turning radius. Sprocket-type turning devices (Fig.



g. 8.9. Arrangement of trolley along hauling chain of conveyer

11*a*) are employed with detachable, minated, double-hinged and other ty- s of chain; with detachable chains, ainly in conveyer sections with a high ain tension (more than 50% of the lowable value).

Turning sprockets usually have 6-13 eth and the base circle diameter of 0-1300 mm; they are cast of steel T (and normalized) or grey iron, or elded of sheet steel Ст. 3.

Turning sheaves (Fig. 8.11*b*) differ the cross-sectional shape of the rim ig. 8.12). Flat-rim sheaves (Fig. 12*a*) can be used with detachable and minated (roller and rollerless) chains; aped-rim sheaves have a central gro- e to accommodate the internal links a detachable chain (Fig. 8.12*b*) or ks of a round-link chain (Fig. 8.12*c*). at-rim sheaves can be used with de- chable chains only at a low chain ten- on (not more than 50% of the allow- le one), since otherwise the inter- l links which are not sufficiently stiff transversely can be deformed easily on a sheave.

Turning sheaves have a diameter of 300-1200 mm; as sprockets, they are cast of grey iron or welded of steel. Welded sheaves may have a diameter up to 1.6-2m.

If the flat rim of a sheave (or rollers in a roller set) is provided with a concentric central recess (Fig. 8.12*b*) 2-3 mm deep and 3-4 mm wider than the internal links of a detachable chain, so that the chain will contact the rim only by external links and the internal links will not be bent, the sheave (or roller set) can be employed at the full allowable tension of the chain.

A sprocket or sheave usually rotates on two rolling bearings (ball, roller or tapered roller) which are mounted on a stationary shaft fastened in a support (see Fig. 8.11). The support is attached to the supporting structure of the conveyer. For proper centering, the support has a lug which enters the centering ho-

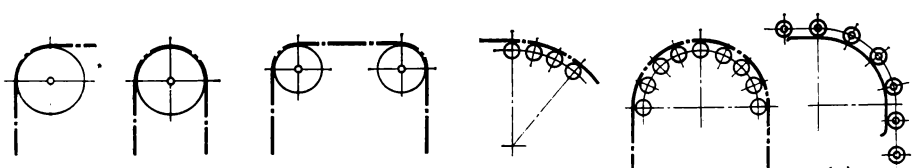


Fig. 8.10. Schematics of horizontal turns in overhead conveyers
a – c, and (c) on sprockets and sheaves; (*d*) and (*e*) on roller set; (*f*) on guide rail

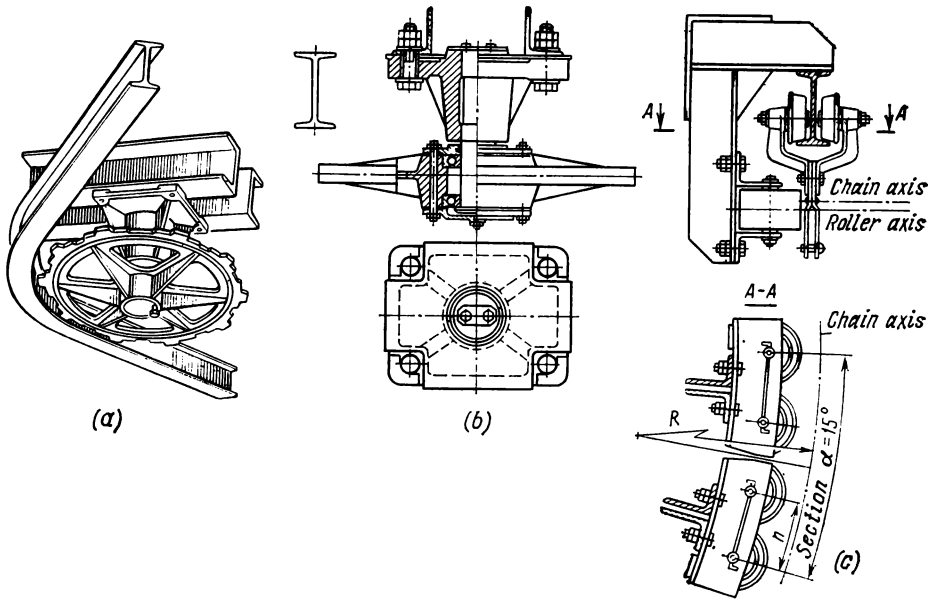


Fig. 8.11. Turning devices
(*a*) sprocket; (*b*) sheave; (*c*) roller set

les in the supporting plate. In modern designs, turning devices are manufactured and mounted together with the curved elements of an overhead track as assembly units of conveyers. This ensures higher assembly accuracy of the curved sections (as regards their coaxiality and levelling of the chain and track) and easy replacement of assembly units in repairs.

A roller set is essentially a number of stationary rollers arranged along an arc, on which chain links can roll (see Fig. 8.11*c*). The shafts of rollers are mounted in a fixed frame attached by brackets to the supporting structure or directly to the overhead track of a conveyer.

Each roller in a set is mounted on two ball or roller bearings and has a diameter of 70 mm; the pitch of rollers can range from 90 mm to 170 mm. Rollers are made of grey iron by casting in metal moulds (in order to obtain high hardness in the rim and thus increase wear resistance) or steel 40 with surface hardening to 42-50 HRC.

Modern conveyer designs employ rollers with two built-in ball bearings, so that the roller rim and hub serve as the ring of a bearing. Roller sets are usually manufactured in sections with a turn angle of 15°, 30°, or 45°, which are joined together to obtain the required angle (up to 180°). The radii of roller sets, as measured to the axis of the hauling chain bending on them, are chosen from

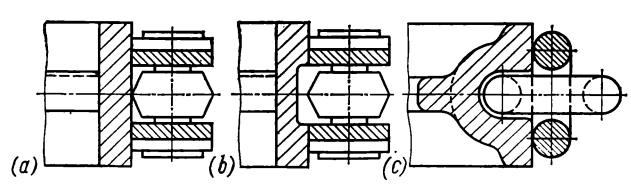


Fig. 8.12. Rim profiles of sheaves

the standard series of 1.0, 1.25, 1.6, 2.0 or 2.5 m.

Roller sets are applicable with detachable and laminated chains with turning angles up to 45°, for larger angles (up to 180°), when it is needed to have a large turning radius (more than 650 mm) in the horizontal plane, since sprockets or sheaves would be too heavy in that case. In conveyers with detachable chains, roller sets and flat-rim sheaves are employed only in path sections with a low chain tension (roughly up to 50% of the allowable value), since the internal links of these chains are poorly resisting to lateral bending.

Guide rails are circularly curved sections of the track on which the wheels of a double-hinged or sectionalized chain can revolve. Turning devices with guide rails are simple in design, but are applicable only with wheel-type tractive elements (see Ch. 3). Guide rails are employed for turning angles up to 180° over a radius (to the chain axis) of 0.63, 0.8, 1.0, 1.25, 1.6, 2.0 or 2.5 m.

The radius of a turning device is selected so as to allow free passage of load-carrying attachments in a bend and depends on the overall dimensions of the attachments and loads and the spacing between them. A graphical construction can be made to check whether the largest load can pass through a bend. The minimal spacing between the protruding portions of loads or attachments should be not less than 100-150 mm. At higher speeds of conveyer and greater height of load-carrying attachments, the spacing should be greater so as to avoid collisions of loads or attachments at conveyer start-up, their occasional tilting, or under the action of centrifugal forces in track bends. This spacing also depends on the kind of loads: fragile or brittle loads require a larger spacing. With loads from 3 to 12 m long (such as wood logs, timber, automobile chassis, reinforced concrete plates, etc.), the turning radii and spacings between parallel conveyer runs become quite large even if their width is not too large. In such cases, special devices are employed which can ensure plane-parallel motion of long loads in turns without their rotation.

Turning devices are mounted on metal stands with cantilever beams, brackets or special fastenings, or suspended either on ties or rigid frames from the upper structures of a building.

Overhead track. The track of overhead trolley conveyers is made of one or two rolled or bent sections of steels Ст. 3, Ст. 5, 45, 09Г2 or 14Г2; the last steel grade ensures a long service life of tracks at heavy loads. Single-beam (monorail) tracks (Fig. 8.13*a-c*) are made from I-beams Nos. 10, 12, 14 or 16 or steel tubes with a diameter of 60, 100 or 150 mm, having a longitudinal cut along the tube length to allow the motion of trolleys, or else from special channel sections. Double-beam tracks (Fig. 8.13*d* and *e*) are made from two rolled or bent angles or two special bent sections.

I-beam tracks have found wide application for medium- and heavy-duty conveyers. Among their advantages are easy manufacture, assembling and fastening, appropriate rigidity of the track, and the possibility for using the upper flanges as counter-rails.

Angles and shaped bent sections are mainly employed in tracks of very light, light and medium-duty conveyers with double-hinged and round-link chains. They have a low mass, can be used with trolleys on cylindrical wheels, and allow vertical bends of a small radius. Among their disadvantages are a low stiffness and much labour required for assembling, fastening, and mounting of counter-rails. Single-profile tracks are preferable over those consisting of two shaped sections.

Overhead tracks are manufactured in individual sections ready for assembly. The joints between sections may be fixed (rigid) or expanding. Track sections are connected by means of centering straps connected by bolts or welding. Since the joints have a smaller

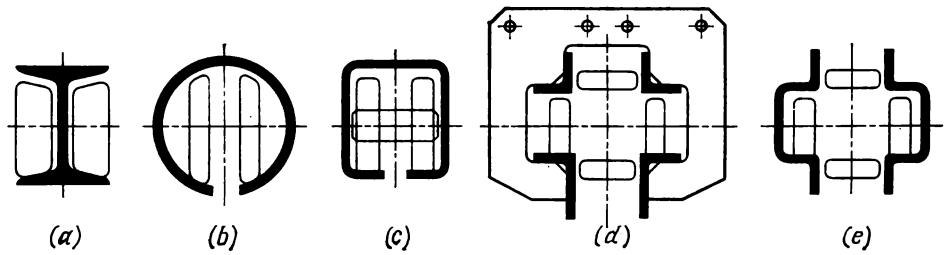


Fig. 8.13. Cross-sectional profiles of tracks of overhead trolley conveyers

strength than the track sections, they should be provided at distances not more than 1-1.5 m from vertical stands. Expanding joints are provided only in sections near the take-up device (the travel in a joint should be 100-400 mm) and also as temperature compensators in conveyer tracks passing through drying ovens or cooling chambers (their expansion length should be 15-20 mm).

Track fastenings should be designed so as to utilize as much as possible the structures of the building where the conveyer will be installed, and to minimize the number of special conveyer stands which will block up the premises. In view of this, overhead tracks are usually fastened on round ties from 16 to 20 mm in diameter (see Fig. 8.14*a*), or rigid suspensions to the ceiling trusses of a building or on brackets to the walls and columns (Fig. 8.14*b*), or else on special stands (Fig. 8.14*c*) when there is no possibility to utilize the structural elements of the building or when the track is not high

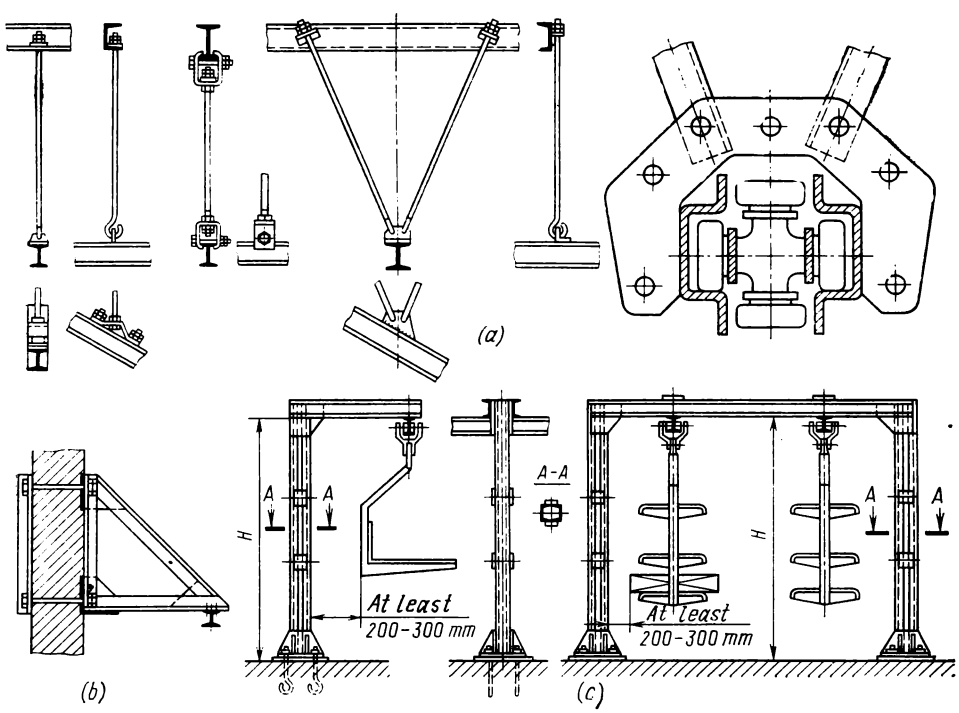


Fig. 8.14. Supporting metal structures of overhead tracks
(*a*) with ties; (*b*) with brackets; (*c*) with stands

enough (up to 2-2.5 m) above the floor.

The bending radii of overhead tracks in the horizontal plane depend on the type and size of turning devices and tractive elements. The bending radius of a track on a sprocket should be smaller than the base circle radius of the sprocket, since a chain link with an attached trolley is arranged along a chord to that circle when bending around the sprocket. For that reason, the bending radius of a track is determined by the centres of joints by which the trolleys are fastened to the chain.

The bending radii on smooth-rim sheaves are determined in a similar manner, i.e. by considering the position of the tractive element (chain) and trolley on a sheave.

Vertical bends of a conveyer (i.e. its curved sections in the vertical plane) are formed directly by bending the chain and track by a selected radius. The radius of vertical bends of overhead conveyers determines the following main factors:

1. *The type, design and pitch of chains.* Double-hinged and round-link chains (which allow a large angle of bending between adjacent links), as well steel cables, may have an appreciably smaller bending radius than detachable chains, for example. For chains of any type, a larger link pitch requires a greater radius of bending.

2. *The profile of an overhead guide track.* Under comparable conditions, a track consisting of two shaped sections (say, of two angles) can have a smaller bending radius, than a single-beam track (say, an I-beam track). With a small bending radius, the chain can be passed between the two track sections, while in a single-beam track the chain must be located at a certain minimum distance from the bottom flange so as not to touch this in motion.

3. *The spacing between trolleys.* A greater pitch of the trolleys which support the chain and load-carrying attachments, requires a larger bending radius. This condition is satisfied by making a chain so that it is more flexible in places of fastening of trolleys and less so in sections between the trolleys which can pass over chords of the bending arc.

4. *The connection of trolleys to chains.* A hinged connection allows turning of the chain links at a trolley by an angle of 20-45°, while rigid connections allow an appreciably lower angle (up to 3-8°). For that reason, chains with hinged fastening of trolleys can bend over smaller radii than those with rigidly fastened trolleys.

5. *The tension of chains, especially of those fastened rigidly with trolleys.* With a higher tension of the chain in a bend (i.e. when the bend is closer in the running-on chain section to the drive), a larger bending radius should be selected so as to minimize stresses and wear of the chain and trolley elements. The recommended radii of vertical bends for overhead conveyers with detachable chains are given in Table 8.2.

If trolleys are mounted on a conveyer with different spacings, the bending radius should be selected for the largest trolley pitch. It is advisable to select a single (the largest) bending radius among the recommended values for all vertical bends of a conveyer, so as to minimize the forces acting on the trolleys and tracks, and thus increase the service life of tracks, chains, and trolleys.

Table 8.2. Recommended Radii R (m) of Vertical Bends for Detachable-chain Trolley Conveyers (to Centre Line of I-beam in Fig. 8.15*b*)

Chain pitch t, mm	Chain tension in a bend (% of allowable)								
	up to 50			75			100		
	with trolley pitch t_t equal to (t=chain pitch)								
	$4t$	$6t$	$8t$	$4t$	$6t$	$8t$	$4t$	$6t$	$8t$
80	1.6	2.0	2.5	2.0	2.5	3.15	2.5	3.15	4.0
100	2.5	3.15	4.0	2.5	3.15	4.0	4.0	4.0	5.0
160	4.0	5.0	6.3	5.0	5.0	6.3	5.0	6.3	8.0

Note. Standard detachable chains with cylindrical wheels. The tilting angle of links $\varphi_{min} = 3°$.

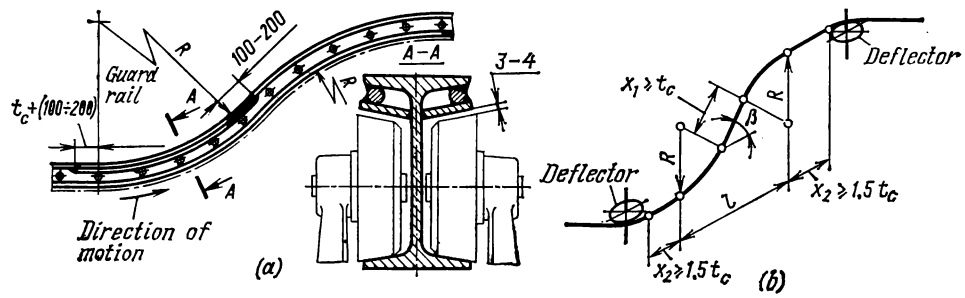


Fig. 8.15. Vertical bend of overhead track
(a) arrangement of counter-rails; (b) schematic of connection with horizontal turns

For double-hinged chains and steel cables, the radii of vertical bends are chosen from the following standard series: 0.6, 1.0, 1.25, 1.6, and 2.0 m. It should be observed that the tractive element does not contact the bottom flange of the track. With tractive elements of these types, vertical bends of a conveyer can also be made with turning sheaves, sprockets, or guide rails arranged in the vertical plane.

When a horizontal turn is combined with a vertical bend, it is required to provide straight sections X_1 and X_2 between the beginning and end points of both and between arcs of the vertical bend; the lowest lengths of these straight sections are indicated in Fig. 8.15. This method ensures that the chain will run properly on and off a turning sprocket or sheave and prevents bending of chain links simultaneously in two directions.

In concave curves of vertical bends (inlets of ascending sections and outlets of descending sections), the tension of the chain tends to lift the trolleys above the track. In order to avoid excessive lifting of trolleys and to guide them properly, such concave curves are provided with guide counter-rails. In I-beam tracks, a guide counter-rail is usually made in the form of a strip (Fig. 8.15*a*) welded at the same angle of incline as that of the beam flange or a longitudinal cut is made in the beam leg and the top flange is lowered (upset). In tracks made from two angles, the counter-rail is made from two additional angles mounted above the main ones; for higher stiffness of a vertical bend, additional angles are often provided along the entire length of a bend, rather than along its concave portion only.

Overhead tracks are calculated by the methods of structural mechanics for the lateral bending strength and local bending of beam flanges under trolley wheels (the allowable total stress is $[\sigma_b] = 120$ MPa), for the lateral bending strain (the allowable sag is $f_{max} = 1/500$ of the span), and for the stability of the plane bending shape. Bent shaped sections are additionally calculated for the restricted (bending) twisting which may appear if the trolley wheels are eccentic relative to the vertical axis of the track section passing through the centre of bending.

Drives. Overhead trolley conveyers employ angular or straight (caterpillar) drives with a constant or variable speed. An angular drive can transmit the tractive force by engagement on a sprocket (for detachable, double-hinged and other types of chains) or knuckles (for sectionalized chains) or by friction (friction drives for steel cables and round-link chains). An angular sprocket drive is usually mounted in turns of the conveyer path through 90° or 180°, though a smaller angle of wrapping of chains on sprockets is principally possible.

An angular sprocket drive (Fig. 8.16)

Table 8.3. Main Characteristics of Reducer Gears Made in the USSR

Characteristic	Reducer gear type			
	КДВ-160	КДВ-200 М2	КДВ-250 М2	КДВ-350 М2
Maximum torque on drive sprocket shaft, N m	800	1600	4000	10 000
Range of transmission ratios in 17 embodiments	81-2909	81-2909	81-2909	81-2909
Range of chain speeds (m/min) with the drive motor rotating at:				
1000 rpm	0.4-0.95	0.4-1.14	0.64-1.76	0.82-1.83
1500 rpm	0.7-22.4	0.6-23.2	0.96-22.8	0.92-23.2
Maximum cantilever load on low-speed shaft, kN	5	9	14	32
Total mass, kg	145	250	360	1250
Total rated efficiency	0.92	0.92	0.92	0.92
Type of trolley conveyer	ЦПК-80Р	ЦПК-80Р ЦПК-100Р	ЦПК-100Р ЦПК-160Р	ЦПК-160Р

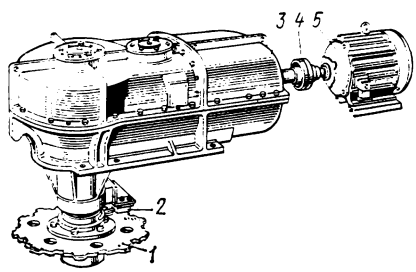


Fig. 8.16. Angular reducer-type drive for overhead conveyers

comprises a drive sprocket *1* mounted on the vertical shaft of a reducer *3*, an electric motor *5* connected with the input shaft of the reducer by an elastic coupling *4*, and a safeguard *2*.

There are three main schemes of reducer for overhead conveyer drives: (1) with worm and cylindrical gearing (Fig. 8.17*a*); (2) with bevel and cylindrical gearing (Fig. 8.17*b*); and (3) with cylindrical gearing and a vertically mounted flange motor (Fig. 8.17*c*).

Worm-gear reducers are quite compact and have a relatively low mass even at high transmission ratios. Among their drawbacks are a low efficiency (up to 0.44), which limits their application. The most popular standard reducers are made by the second scheme (Fig. 8.17*b*) and their parameters are given in Table 8.3.

Reducers with a vertical flange motor have a compact design and high efficiency; they are applicable with light-type overhead trolley conveyers. The electric motor is connected with

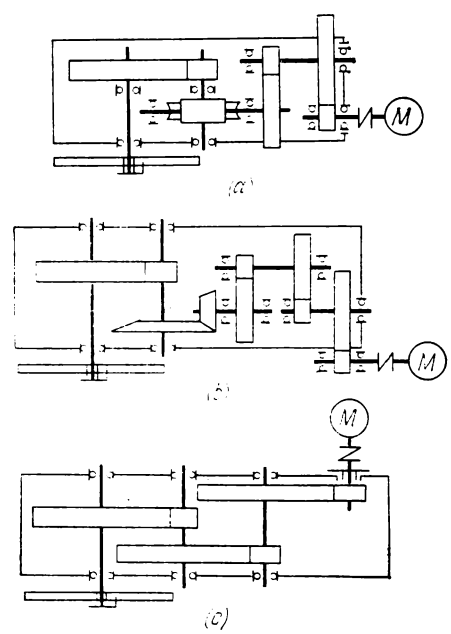


Fig. 8.17. Kinematic diagrams of reducer-type drives

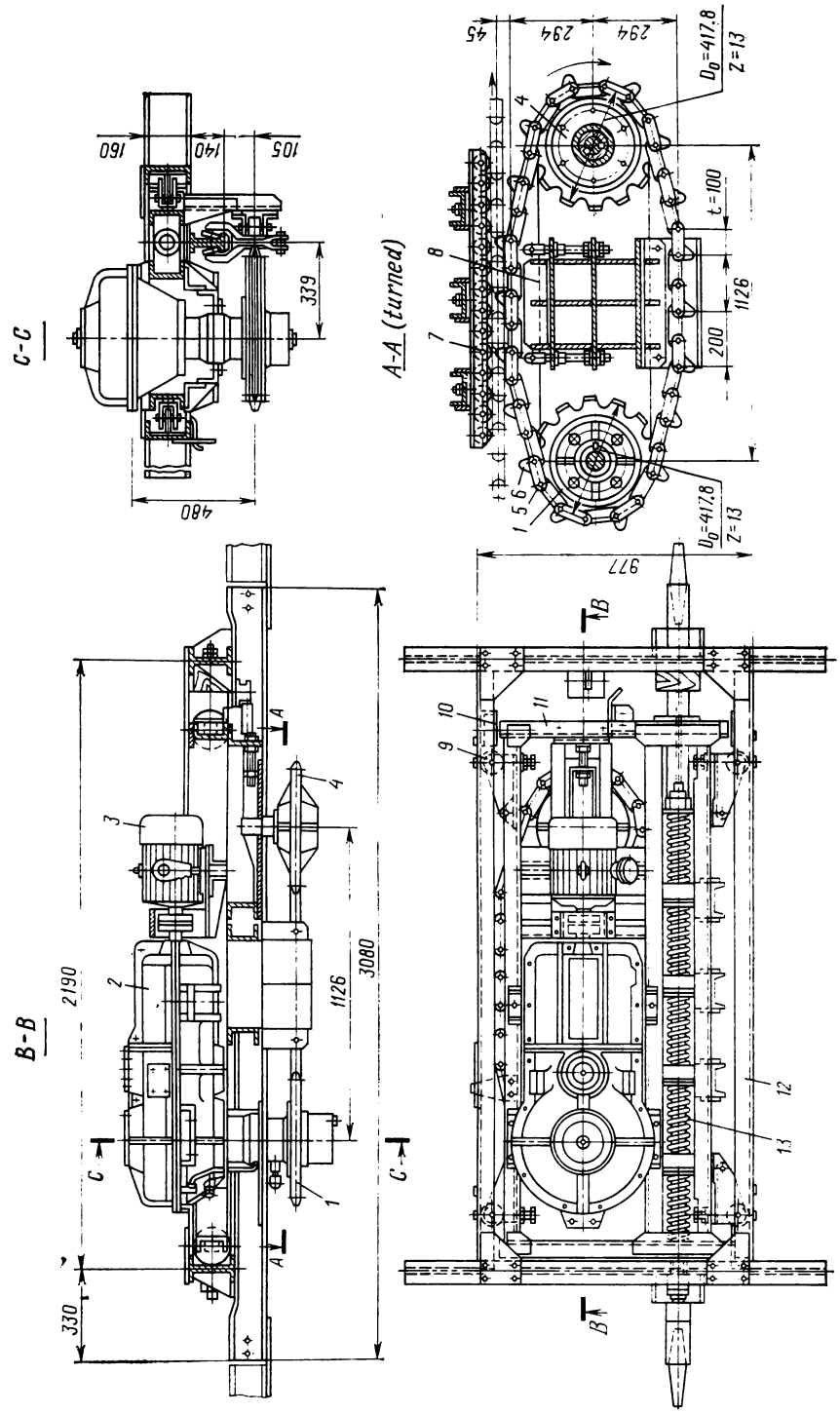


Fig. 8.18. Caterpillar drive of conveyer type ЦПК-400Р

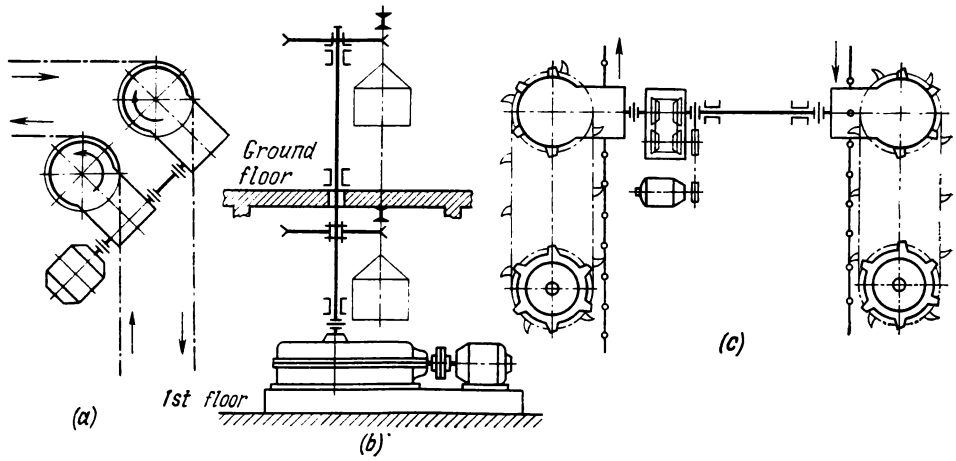


Fig. 8.19. Schematics of combined drives
(a) angular drive with horizontal shaft; (b) angular drive with vertical shaft; (c) caterpillar drive with horizontal shaft

the reducer through an elastic coupling (mostly with rubber-bushed studs) or a hydraulic coupling; V-belt transmission is sometimes used in variable-speed reducers.

A caterpillar drive (Fig. 8.18) has a drive sprocket *1* and take-up sprocket *4* around which moves a driving (caterpillar) chain *5* with knuckles *6*; the drive also has a guide rail *8* and roller set *7*. The drive sprocket is rotated by an electric motor *3* through reducer *2*. A caterpillar drive is mounted in the horizontal plane on a straight section of the conveyer path and the tractive force is transmitted by the knuckles of the caterpiller chain which engage the joints of the conveyer chain.

A guide rail and a roller set ensure straight motion of the chains and do not allow their deflection from each other. The caterpillar chain is made as a special-type laminated roller chain. Knuckles of the chain are made of steel 45 or 40X and their surface is heat-treated to a hardness of 44-48 HRC. The diameters of drive sprockets are usually taken within the range of 400-500 mm and the span between them, 0.9-1.2 m.

The mechanisms of a caterpillar drive are mounted on a movable frame *11* having support wheels *10* and guide wheels *9* and a spring-biased stop *13*. The frame can move on the wheels inside a stationary frame *12*. In the operation of the drive, the force that is directed opposite to the tractive force of the caterpillar chain compresses the springs and moves the internal frame. The travel of this frame corresponds to the tractive force, so that as the pressure exceeds the allowable value, the frame engages a limit switch and the conveyer is stopped. The frame motion is indicated on the scale of a dynamometer. An elastic fastening of the frame is favourable for minimizing the dynamic forces at conveyer start-up.

Angular and caterpillar drives have been compared in Chapter 3. Of the two types, caterpillar drives are more popular.

Combined drive. In modern large works, overhead trolley conveyers handling various materials in and between shops must have an appreciable length (2-5 km or even more) and a high load-carrying capacity. With long conveyers, single drives are inapplicable even with a low mass of transported goods, since the chain tensions exceed substantially the allowable limits for standard chains. The problem cannot be solved rationally by arranging a number of suc-

cessive conveyers, since this would interrupt the continuous conveying process and require additional reloading stations, and therefore, involve labour and time losses, which is inefficient or sometimes inadmissible.

Continuous conveying of goods to large distances can be effected most rationally by a conveyer having a combined drive or, most often, by a multiple-drive conveyer. A combined drive has two or, less frequently, three drive sprockets driven from a common transmission mechanism (a single mechanical shaft) from a single electric motor (Fig. 8.19). This solution makes it possible to divide the entire conveyer path into two or three portions and thus diminish the chain tension.

A combined drive can be simple in design only when a number (two or three) of carrying runs of a conveyer pass close to one another at the same level in a horizontal plane (Fig. 8.19), or one under another in a vertical plane, with chain tensions in these runs being roughly equal. Such favourable conditions appear, however, only rarely, mainly in heavy-duty conveyers of horizontal driers, test stations, and cooling conveyers of a length of 300-500 m.

Multiple-drive conveyers. Multiple drive of a conveyer is the most common and rational method for solving the problem of load continuous conveying to large distances with the use of standard chains. A multiple-drive conveyer (Fig. 8.20) has a number of drive mechanisms (from 2 to 12) arranged at suitable points of the conveyer path and operating in a matched mode. In modern practice, multiple-drive conveyers are known to have the path length up to 3.5 km. There are a number of conveying systems with overhead trolley conveyers in which the operation of the drives is synchronized electrically.

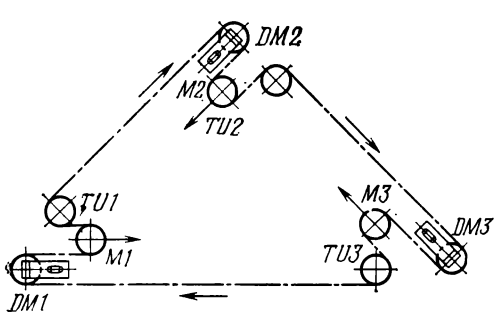


Fig. 8.20. Schematic of overhead conveyer with three-motor drive
DM—drive mechanism; *M*—motor; *TU*—take-up

In this country, the widest application has been gained by a simple and reliable system of cooperating driving mechanisms in which a constant speed of the chains is controlled without electrical synchronization. In this system, all drive motors operate conjointly and the required tractive force for the conveyer sections is divided evenly between them. The drive motors are connected with one another only by the hauling chain of a conveyer. The chain effects their synchronization and for that reason is subjected to an additional tension, which increases with a greater load nonuniformity of the individual conveyer sections between the drive mechanisms and with the number of drives. The additional tension of the chain which appears at uneven loading is a certain drawback of the system. Uneven loading of some sections of a conveyer, even with a uniform distribution of the drive mechanisms over the conveyer path, may appear during loading or unloading of the conveyer, or on sudden interruptions in load supply. In such short time intervals when some sections of a conveyer run loaded and the others are empty, it is essential to diminish the force acting on the most heavily loaded electric motor, and therefore, to apply an additional tension to the chain.

In the system described, all drive mechanisms of a conveyer must have the same strictly constant transmission ratio, and the electric motors must be of the same type and have the same characteristics of power, rotational frequency and torque. They must have an increased slip (6-10%), which should

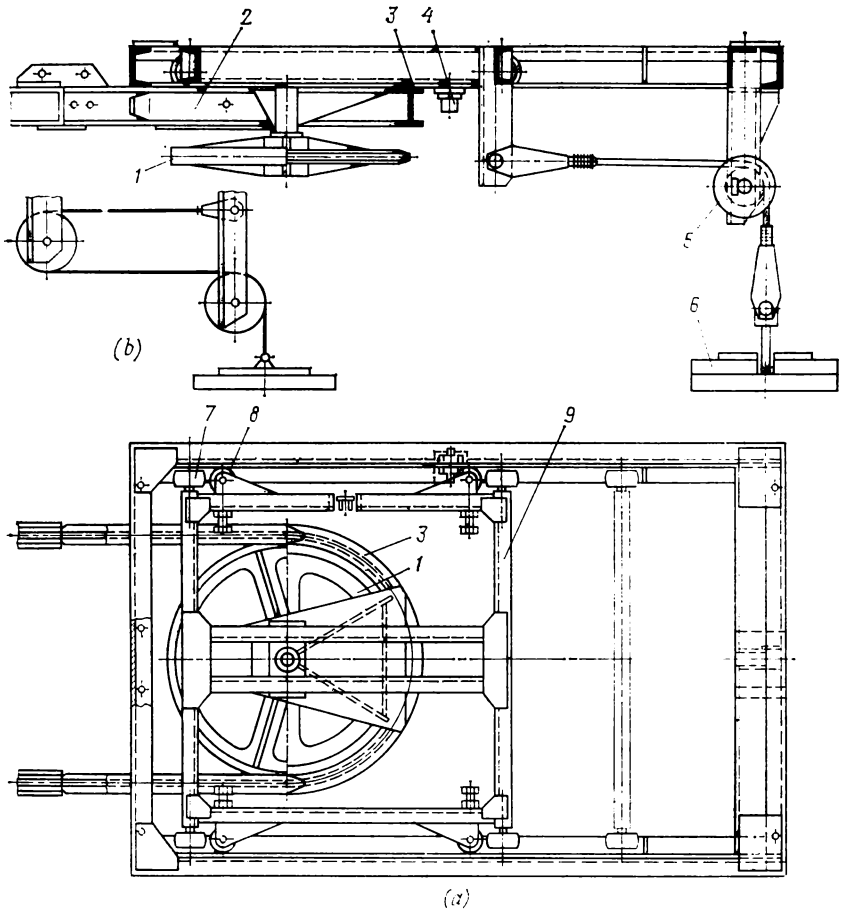


Fig. 8.21. Gravity-type take-ups
(*a*) without tackle block; (*b*) with tackle block

be the higher, the greater is the probable unevenness of loading. For that reason, such systems employ asynchronous phase-rotor motors with an additional resistance in the rotor circuit, electric motors type AOC which have an elevated slip, or common squirrel-cage motors in combination with hydraulic couplings. Hydraulic couplings provide the required slip and ensure uniform distribution of the load between the motors of the conveyer drive.

In this country, multiple-drive conveyers with two to five jointly operating drive mechanisms are employed successfully.

Conveyer drives with electric synchronization (for instance, by the 'electric shaft' method) have found no wide application in view of their intricate design and some difficulties in operation.

Take-up devices. Overhead trolley conveyers employ gravity, pneumatic, hydraulic, spring-screw and screw-type take-ups. A gravity take-up (Fig. 8.21) has a carriage *9* with a turning device *1* (turning sprocket, sheave or roller set), a moving curved section *3* of the conveyer path, and a take-up weight *6* with cables and guide sheaves *5*. Under the action of the weight, the car-

riage moves backwards, stretches the chain which runs around the turning device and thus provides the required initial tension of the chain. The total travel of the take-up carriage (take-up travel) is usually 250, 400 or 600 mm.

The take-up carriage is mounted on four vertical supporting *7* and horizontal guide wheels *8*. The moving section *3* of the conveyer path, which is mounted on the carriage, is connected with the fixed ends of tracks by means of expanding joints *2*. An expanding joint has an upper and a lower guide which ensure continuous connection of the fixed and moving portions of the track at different travel of the take-up carriage. The take-up device has limit switches at the extreme points of carriage travel.

Gravity-type take-ups are made either without a tackle block (Fig. 8.21*a*) if the tension is less than 4 kN or with a tackle block (Fig. 8.21*b*) for tensions up to 16 kN.

Gravity-type take-ups have found wide application in overhead trolley conveyers due to their automatic action and high reliability. They are always employed in conveyers passing through drying, heating or cooling chambers, in conveyers with friction-type and combined drives, and in multiple-drive conveyers. The number of take-ups in combined-drive conveyers should be equal to the number of drive sprockets; otherwise, overtensioning and uneven wear of the chains is likely to occur. In multiple-drive conveyers, the number of take-ups is equal to the number of drive mechanisms. The tensioning force and mass of a take-up weight can be determined by formulae (3.18) and (3.19).

In light-type conveyers (with the chain tension up to 4 kN), the take-up and drive are sometimes combined by mounting the drive motor on the take-up carriage. This simplifies the conveyer path and makes it possible to eliminate two or three turning devices, but requires a higher tensioning force. At high tensions of the chain (say, more than 4 kN), combined arrangement of the drive and take-up is inefficient in view of inevitable skewing of the take-up carriage (owing to the high difference in chain tensions in the running-on and running-off sections).

Load-carrying attachments. Load-carrying attachments of overhead trolley conveyers serve to convey loads on supports or in suspended state. Their design may be diverse depending on the kind, dimensions and mass of load units, the type of conveyer, and the method of loading and unloading. Load-carrying attachments are made in the form of shelf stacks, troughs, hooks, lever clamps, boxes, chain ties, strings, etc. (Fig. 8.22, *a-d*). The parameters of load-carrying attachments for unit and packed goods are mostly standardized. Load-carrying attachments should be strong, light, simple and inexpensive in manufacture, convenient for loading and unloading, and ensure safe transportation of load units in inclined and curved, as well as in horizontal sections of a conveyer. Load-carrying attachments are hinged to the trolleys so that they are always in the upright position during the transportation.

Trolley conveyers with double-hinged chains or with hinged attachment of the tractive element to trolleys may have vertical ascending and descending sections. In such conveyers, the load-carrying attachments are fastened to trolleys by means of a cantilever bar and two bars which form a triangle (see Fig. 8.3). The overhang of a cantilever bar should be greater than half the largest overall dimension of the load or attachment, so that the load could not contact the chain in vertical conveyer sections.

Loading and unloading of attachments can be performed manually, by hoisting means, semi-automatically, or automatically.

In semi-automatic loading and unloading, the most labour-consuming operations are carried out by the conveyer proper. This is most often effected in

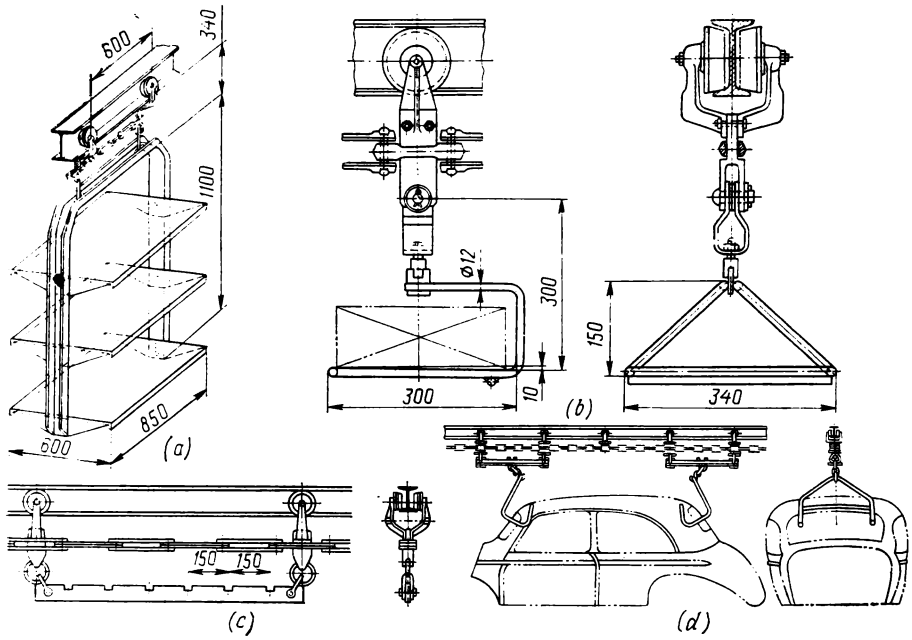


Fig. 8.22. Types of load-carrying attachments

vertical ascending or descending conveyer sections where the loads can be easily held up by or hanged onto the conveyer attachments or taken off (unhooked) from them. For instance, a conveyer may have a steep vertical incline and decline at the loading station and a table beneath the conveyer, onto which load units (boxes, castings, etc.) are placed. The load units on the table are attached manually to the hooks, strings, lever clamps, etc. of the load-carrying attachments which are then transported by the conveyer trolleys along the ascending section.

Unloading of the conveyer is performed in the reverse order: in a descending conveyer section, load units are lowered by the conveyer onto a table and the operator unfastens them from the load-carrying attachments. Conveyer unloading can also be performed in a straight horizontal conveyer section onto a lifting table. The table lifting can be hand- or foot-operated, pneumatic or electric, with manual or automatic control. Automatic loading and unloading systems with automatic addressing of loads find wide application. Systems with loading and unloading on horizontal sections of the conveyer path seem to be most promising.

Automatic loading of load-carrying attachments consists of the following operations. Load units are delivered in a given order to the loading station of a conveyer. As the load-carrying attachments pass over the loading station, they catch up the load units, or a special automatic addressing device switches on the automatic loader which prepares the load units to loading and hangs them up onto the passing load-carrying attachments of the conveyer (Fig. 8.23*a*). Automatic unloading involves the following operations. During loading, the load units are addressed automatically to particular destination points. On arrival to that point, a load-carrying attachment switches automatically an unloading device (or comes into direct contact with it), so that the

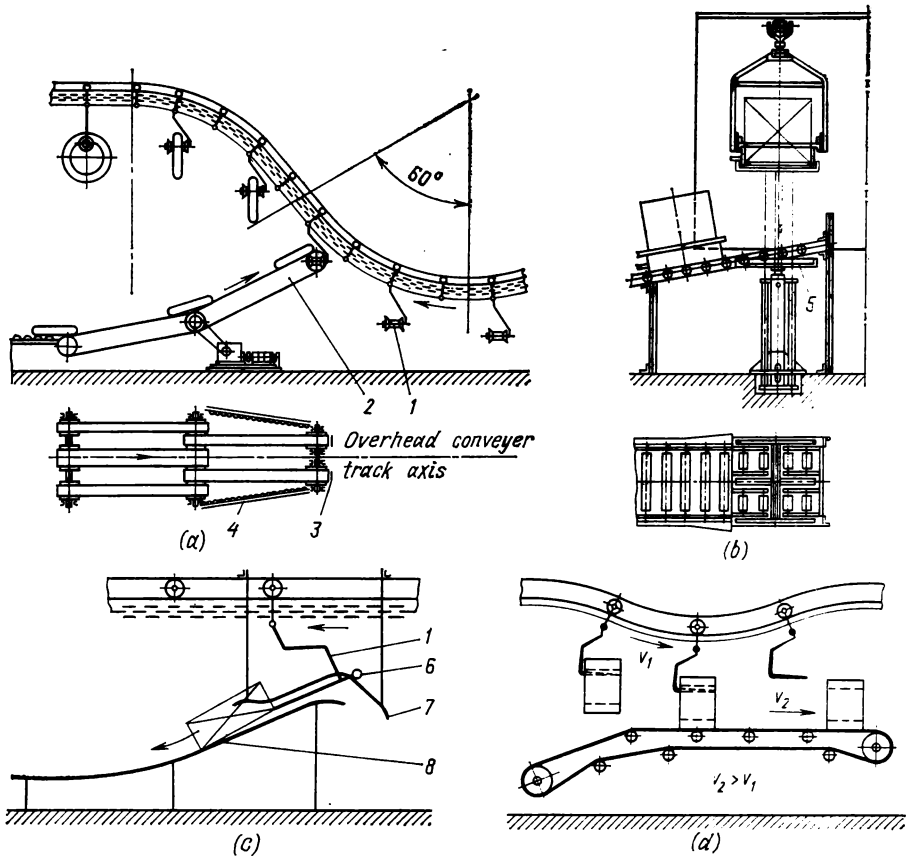


Fig. 8.23. Schematics of automatic loading and unloading

(*a*) loading; (*b*), (c, and (*d*)] unloading; *1*—load-carrying attachment; *2*—loading conveyer; *3*—stop; *4*—guides; *5*—lifting table; *6*—attachment roller; *7*—guide rail; *8*—slide

attachment is tilted, turned, disengaged, etc. and is freed from the load, or the load on the moving conveyer is transferred (or pushed off) onto a stationary or moving receiving table, etc. (Fig. 8.23*b*, *c* and *d*).

Systems of automatic addressing of load-carrying attachments. In automatic addressing systems, load units are automatically loaded and unloaded at particular destination points to which they are addressed. Some systems have only automatic loading or automatic unloading. Automatic addressing is employed most widely in overhead trolley conveyers for distribution of load units between various sections of the conveyer path.

Automatic addressing systems are divided into decentralized (local) and centralized (controlled from a central panel). Combined addressing systems are also known.

Centralized addressing systems are more promising, but more intricate in the design and organization of production process. Decentralized addressing systems are simpler and more easily adaptable to production conditions and their variations, and for that reason are now used more widely.

A decentralized addressing system in any design embodiment generally comprises (Fig. 8.24) an addressing means, i.e. an address carrier with a set of addressing elements, a reader to

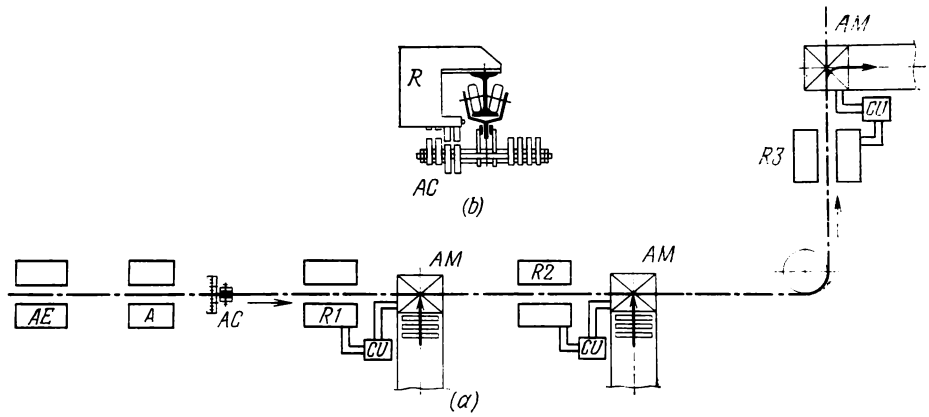


Fig. 8.24. Addressing systems
(a) decentralized; (b) with contact reading; *AE*—address eraser; *A*—addresser; *AC*—address carrier; *CU*—control unit; *AM*—actuating mechanism; *R*—reader

read off the address, an automatic address eraser (to bring the addressing elements into the initial 'addressless' state), and an automatic addresser which sets up an address. In some cases, such as in programmed addressing, addressers and address erasers are not mounted on the conveyer path.

Addressing elements may be made in the form of discs, pins, keys, lugs, contacts, etc. Their combination on an address carrier determines in an adopted code a particular destination address of the load or a particular operation that should be performed with the load.

An address carrier *AC* is mounted on a load-carrying attachment of conveyer and determines the destination point (address) or the path of motion of the load. Readers *R* are provided at particular points of the conveyer path before loading and unloading stations (Fig. 8.24*a*). As a load-carrying attachment with an address carrier passes through a reader, the latter decodes the address set up in the address carrier. If the position of the addressing elements in the address carrier coincides with the position of the receiving elements (probes, levers, lights, etc.) of the reader, i.e. the address set up on the attachment is coincident with that of the reader (Fig. 8.24*b*), the latter sends a command into a control unit *CU* of the loading or unloading mechanism and this switches on an actuating mechanism *AM*. Address erasers *AE* are mounted after the unloading points at the beginning of a new route. An automatic addresser *A* is arranged at the beginning of a new route after an address eraser and, as a rule, after an unloading station at a destination point.

In decentralized addressing systems, an address of a load is set up by the operator during loading of the load-carrying attachments. The operator sets up manually the addressing elements on the attachment (keys, pins, etc.) or presses the required button on an addressing panel; in the latter case, an address on a load-carrying attachment is set up automatically by the addresser when the load-carrying attachment passes through it.

Address carriers can be arranged on load-carrying attachments (or trolleys) either horizontally (Fig. 8.25*a*) or vertically (Fig. 8.25*b*). Horizontal arrangement of address carriers is more preferable, since it requires a smaller cross-sectional area of the conveyer (in the vertical plane). With long horizontal address carriers, however, their

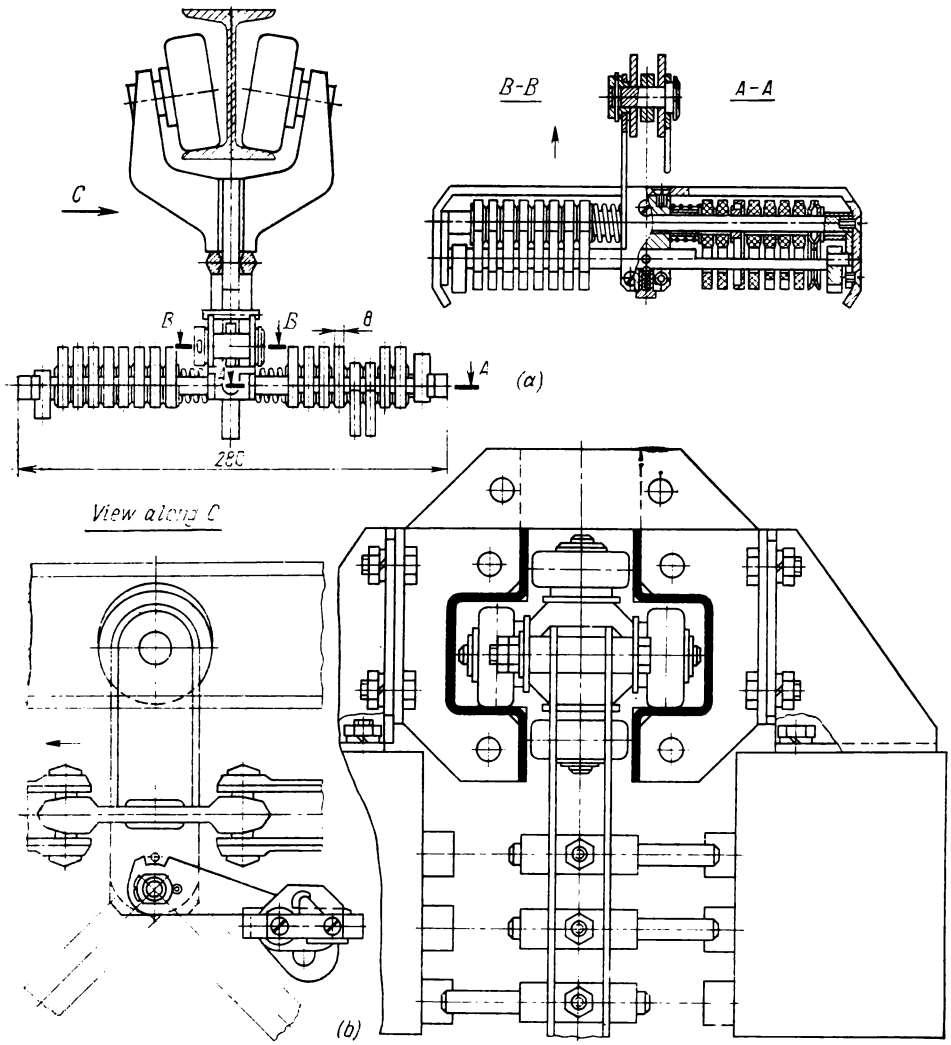


Fig. 8.25. Address carriers
(a) horizontal; (b) vertical

skewing is likely to occur due to load-carrying attachment skewing or under the action of the moment appearing on contact reading-off of the addressing elements.

Among the drawbacks of vertical address carriers are greater dimensions along the conveyer height and probable sideway oscillations which can cause errors in addressing. On the other hand, they are simple in design and can be easily fastened on load-carrying attachments. For greater reliability of operation, address carriers of any type should have as low a length as possible (usually not more than 300 mm).

Address carriers can be mounted either on the load-carrying attachments or on a bracket attached directly to the bracket of a trolley before and above the hinged joint of the load-carrying attachment. In the latter version, the

address carrier has smaller vertical dimensions and steadier position in motion.

Address carriers and readers should have self-adjusting ('floating') fastening to ensure their appropriate interaction at the allowable manufacturing and mounting tolerances of the conveyer. Depending on the kind of interaction of addressing elements, address reading may be of contact type or contactless.

Contact methods of address reading include electrical [closing of low-voltage (12-36 V) contacts], mechanical (interaction of cams, discs and like elements with deflecting bars, levers, etc.), electromechanical (interaction of pins, stops, lugs and the like with limit switches which energize or de-energize an electric circuit), pneumomechanical (interaction of pins, rollers or levers with pneumatic control elements for conveying dusty or explosive materials).

Contactless reading-off methods are divided into photoelectric (the effect of light rays on photo-cells), inductive (based on variations of a magnetic flux or an e.m.f. induction in electromagnet windings), radioactive (based on radiation of isotopes), magnetic (effect of magnets on switches), etc.

Systems with contactless reading of addresses are more effective, since their addressing elements are not subject to wear, but are more intricate in design.

The capacity of an address carrier, i.e. the number of probable addresses, is determined by the number of its addressing elements (usually not more than 16), their arrangement, the method of reading (single- or two-sided), and the number m of the elements (one, two, three, or four) which can be set up simultaneously. Thus, the number of possible addresses A is determined as the combination of m elements from their total number n. For the most usual single-sided reading-off of addressing elements

$$A = C_n^m = \frac{n\,(n-1)\,(n-2)\,\ldots\,[n-(m-1)]}{1\times 2\times 3\,\ldots\,m}$$

With centralized addressing, load-carrying attachments of a conveyer have no address carriers, and the destinations are set up from a central panel by a programme of load transfer between technological processes and stores. In the simplest case, the route for a load is set up by the control panel operator. In more intricate systems, the route of load-carrying attachments is written by a special code as a programme on perforated cards or tape, magnetic tape or another programme carrier. The latter is mounted on the central control panel for conveyer operation where it is read off to determine the routes of loads. In centralized addressing, use is also made of various tracing (follow-up) devices, such as electromechanical addresses with a rotating 'programme drum', electric devices with a set of logical elements, etc.

Actuating mechanisms for loading and unloading of conveyer attachments at addressed destination points may be in the form of lifting tables (pneumatic, hydraulic or electromechanical), tilting clamps, movable bars, etc.

Safeguards. In order to protect the drive mechanism and moving parts of a conveyer against damage on occasional overloading or on jamming of a load-carrying attachment in fixed structures, the drive sprocket of a conveyer has a shear pin which is cut off as the tractive force exceeds the rated value by 25% (see Ch. 3); a special device then de-energized the drive motor and the conveyer is stopped.

All ascending and descending sections of a conveyer and the horizontal sections above passages are provided with troughs or like protections to avoid falling of conveyed loads on the floor. They are usually made from wire net or steel sheets fastened to the conveyer structure (Fig. 8.26).

On breakage or occasional detachment of the hauling chain, the trolleys with load-carrying attachments on inclined or declined conveyer sections

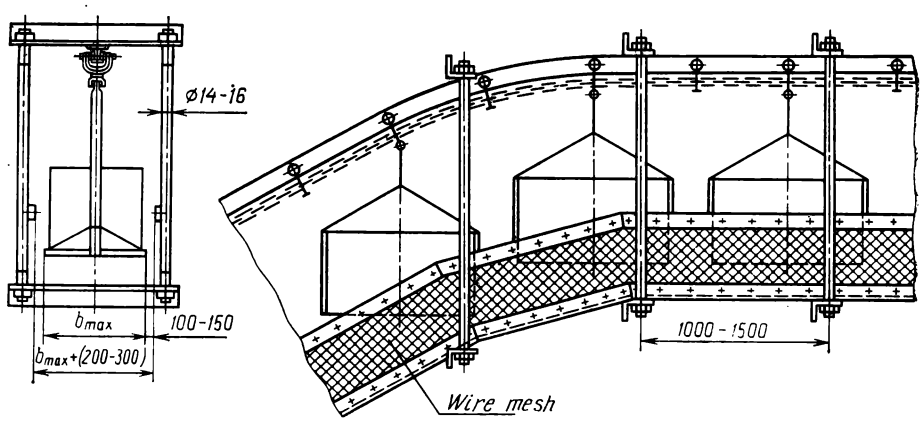


Fig. 8.26. Overhead conveyer enclosure

will tend to run downwards under the action of gravity, which might result in damage of the loads and conveyer. To prevent this, inclined conveyer sections of a lifting height more than 1 m are provided with catchers to catch and stop a broken chain. Catchers are diverse in design depending on their operating principle (ratchet, lever-type, electromechanical, etc.), their position in the conveyer path (in ascending or descending sections), the type of tractive element, and the profile of conveyer track. The simplest catcher employed in ascending conveyer sections (Fig. 8.27) has a steel bracket *1* pivoted on an axle *3*. The axle is fastened in a stand *4* welded to the top flange of the conveyer track I-beam. With normal motion of the conveyer chain (upwards), the bracket is deflected freely by the trolley bracket *2* and then is removed into the initial position by spring *6*. In an emergency situation, as the chain begins to move back, the bracket is deflected rearwards, bears by its cross-piece against the beam flange, and stops the trolley; at the same time, it actuates the lever of a switch *5* (or closes the eyehole of a contactless switch) to de-energize the drive motor

In trolley conveyers passing through painting, sand-blasting, washing

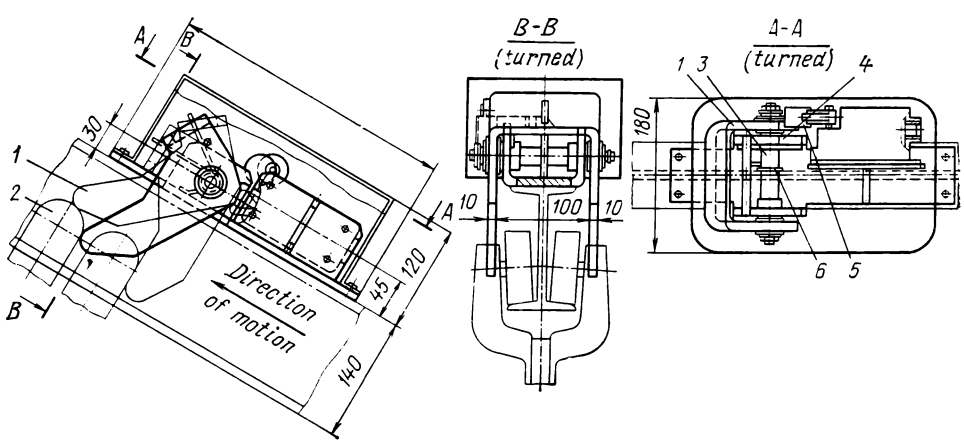


Fig. 8.27. Catcher on ascending conveyer section

or pickling chambers, the moving parts should be preferably closed by a casing with a rubber or asbestos labyrinth seal. The casing is usually connected to a ventilation system and reliably protects the chains, trolleys and track against dirt, overheating and harmful vapours. When a casing will be ineffective or impossible to make, it is better to arrange the track and chains of a trolley conveyer above such chambers, so that only the load-carrying attachments will move through them.

8.2.3. Calculation of Trolley Conveyers

An overhead trolley conveyer is calculated for a given conveying path, the mass and dimensions of load units to be transported, and the required throughput capacity and operating conditions. The principal parameters for the calculation of a conveyer are the throughput capacity Z (units per hour), conveyer speed v (m/min), the pitch of load-carrying attachments a_s (m), and the number of load units on an attachment, i (pieces). These parameters are interrelated by formula (2.14).

The smallest pitch of attachments, $a_{s\,\min}$, should allow free passage of load units of the greatest length $b_{\max}$ through horizontal turns of the smallest radius $R_{\min}$ (Fig. 8.28*a*) and through vertical bends of the highest angle of incline $\beta_{\max}$ (Fig. 8.28*b*). Then we have

$$a_s \geqslant (b_{\max} + \Delta)/\cos \beta_{\max} \qquad (8.11)$$

where $\Delta = 0.15\text{-}0.2$ m is the minimum gap between adjacent attachments with loads.

For conveyers with vertical sections of the path (Fig. 8.28*c*)

$$a_s \geqslant h_{\max} + \Delta \qquad (8.11')$$

The pitch of attachments a_a should be a multiple of two chain pitches.

The speed of overhead trolley conveyers is chosen depending on the mass of loads, the required throughput capacity, and the method of loading and unloading; it is usually within 3-25 m/min and only in rare cases attains 35 m/min. The typical speeds of overhead trolley conveyers are given in Table 8.4. For conveyers employed in process operations, the speed is governed by the rate of a production process, in particular, by the required time for cooling, drying, etc. of conveyed goods on the conveyer (see Ch. 9).

The pull of an overhead trolley conveyer is calculated as for any type of conveyer with a tractive element (see Ch. 2). For proper operation of a conveyer, the lowest chain tension S_0, which is expected to appear in a conveyer section after the most heavily loaded des-

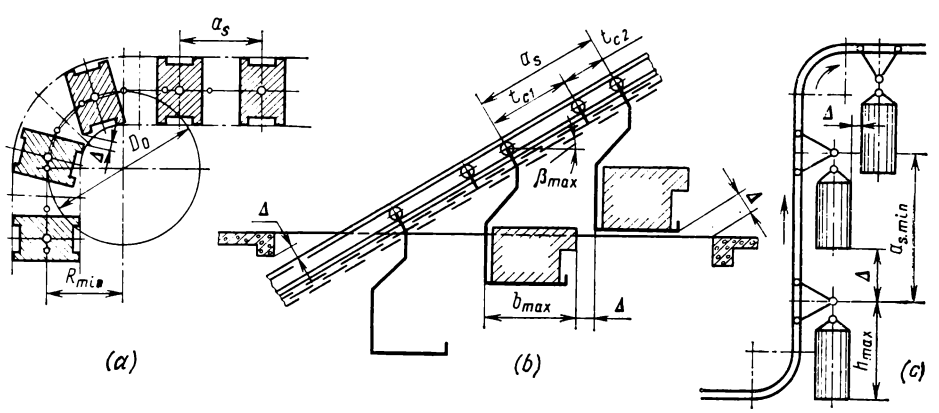


Fig. 8.28. Location of load-carrying attachments
(*a*) in a horizontal turn; (*b*) in an inclined section; (c) in a vertical section

Table 8.4. Speed Characteristics of Trolleys of Overhead Trolley and Pusher Conveyers

Conveyer application	Load mass, kg	Method of loading and unloading	Characteristic speed, m/min
Transporting castings, forgings and assembly units in shops and between them	500-1000	Hoisting means	4
	10-12	Manual	10
	1-5	Manual	12
	50-100	Automatic	18
Transporting machine parts (workpieces) between machine tools	100-500	Hoisting means	10
	5-15	Manual	6
Transporting mould cores to mould assembling	10	Manual	6-4
Transporting rubber articles and pieces between operations	100-200	Hoisting means	10-6
		Automatic	12-6
	10-12	Manual	10
		Automatic	18
Transporting bales, packs and parcels in stores and mail offices	5-10	Manual	12-18
	20-100	Automatic	16-24

cent or at the point where the chain runs on the drive sprocket (in horizontal conveyers), should be not less than 500-1000 N. For friction-drive conveyers, S_0 is calculated by the condition that ensures the transfer of the required tractive force (see Ch. 4).

The rated linear forces (N/m) in the return (empty) and carrying (loaded) run of a conveyer, q_0 and q, can be found by the formulae

$$q_0 \approx g\,(m_a/a_s + m_t/t_c) + q_{ch} \qquad (8.12)$$

$$q \approx q_0 + \frac{gm_l}{a_s} \qquad (8.13)$$

where m_a and m_t are the masses of an attachment and trolley, kg; a_s is the pitch of attachments, m; t_c is the pitch of trolleys, m; q_{ch} is the linear gravity force of the tractive element (chain), N/m; and m_l is the mass of the load on the attachment, kg.

Before calculating the rated linear forces, one has to select the standard size of the tractive element and trolleys and a particular type of load-carrying attachments. The tractive element is chosen by the rated maximum tension S_{max}. For conveyers with the tractive force transmitted by engagement on a sprocket (angular drive) or on knuckles of a drive chain (caterpillar drive), S_{max} can be found approximately by the formula

$$S_{max} = S_0 K_l + w\,(qL_l + q_0 L_e)\,(1 + AK_l) + qH \qquad (8.14)$$

where S_0 is the initial tension of the chain (S_0 = 500-1000 N); $K_l = \varphi^x \xi^y \lambda^z$ is the total coefficient of local resistances to the motion of trolleys [here φ, ξ and λ are the coefficients of resistances respectively in vertical bends, horizontal turns on a sprocket or sheave (ξ) and on a roller set (λ), and x, y, and z are the numbers of vertical bends (x), horizontal turns on sprockets or sheaves (y), and turns on roller sets (z) along the conveyer path]; w is the coefficient of resistance in straight sec-

tions; L_l and L_e are the horizontal projections of the loaded and empty conveyer runs, m; A is a coefficient depending on the number of turns and bends and their positions in the conveyer path, $A = 0.5\text{-}0.3$ (lower values relating to conveyers with more than five turns and bends); and H is the largest height of lift of loads in the conveyer path, m.

Using the tension thus obtained, one can select a standard size of hauling chain by the condition

$$S_{max} \leqslant S_{al}$$

A detailed calculation of a conveyer is carried out by successive summation of the resistances in individual sections of the conveyer path, which can be found by the formulae:

for a horizontal straight section

$$S_n = S_{n-1} + wq'l \qquad (8.15)$$

for a turning sprocket (or sheave)

$$S_n = \xi S_{n-1} \qquad (8.16)$$

for a roller set

$$S_n = \lambda S_{n-1} \qquad (8.17)$$

and for a vertical bend

$$S_n = \varphi\,(\varphi S_{n-1} + wq'l \pm q'h) \qquad (8.18)$$

Table 8.5. Resistance Coefficients w of Trolleys on Ball Bearings in Straight Sections of the Path of Overhead Trolley Conveyers with Detachable Chains and I-beam Tracks

Operating conditions	Mass of load carried by trolley, kg		
	up to 50	100	200 and more
In heated rooms at temperatures above 0°C			
Light	0.020	0.014	0.012
Moderate	0.027	0.020	0.016
Heavy	0.032	0.025	0.020
In unheated premises with winter temperatures down to −20°C; lubrication of bearings by low-temperature greases; for any mass of load			
Moderate	0.038	0.038	0.038
Heavy	0.045	0.045	0.045

where S_n and S_{n-1} is the tension at the end and beginning of a section considered, N; q' is the linear load, N/m ($q' = q$ for the loaded run and $q' = q_0$ for the return run); l is the horizontal projection of the path section considered, m; and h is the difference of levels at the end and beginning of a section, m.

In formula (8.18), the term $q'h$ is taken with a 'plus' sign for ascending sections and with a 'minus' sign for descending ones.

The resistance coefficients w, φ, ξ and λ in formulae (8.14)-(8.18) can be taken from Tables 8.5 and 8.6.

The tractive force on the drive sprocket and the power of the drive motor are found by formulae (2.14) and (2.15) for the highest speed and maximum load. The allowable tensions of detachable chains to ensure their proper durability are given in Table 8.7.

Example 8.1. Calculate an overhead trolley conveyer for transporting of castings from the cut-off department to store.

Initial data. The schematic of the conveyer is shown in Fig. 8.29. The mass of a casting is $m_l = 18.3$ kg; its largest dimensions are 320 × 200 × × 220 mm. Load-carrying attachments will be loaded and unloaded automatically. The rated conveyer capacity: $Z = 600$ units per hour. The conveyer will operate 16 hours per day under moderate conditions.

Calculation. Initially we select a detachable hauling chain type P2-100-220 as the tractive element for the conveyer, with the pitch 100 mm. Considering the dimensions of castings and deciding that they will be transported with the longest side along the conveyer path, we select the pitch of load-carrying attachments: $a_s = 0.6$ m. The highest angle of incline of the conveyer path in a vertical bend is $\beta_{max} = 35°$, and therefore, we have by formula (8.11)

$$a_s \cos \beta_{max} \geqslant b_{max} + \Delta$$

Table 8.6. Resistance Coefficients of Trolleys on Turning Devices and in Vertical Bends of Overhead Trolley Conveyers

Operating conditions	ξ		λ					φ		
	at turning angles, degrees									
	90	180	up to 30	45	60	90	180	up to 20	35	45
In heated rooms at temperatures above 0°C										
Light	1.020	1.030	1.020	1.025	1.030	1.040	1.07	1.010	1.015	1.025
Moderate	1.025	1.035	1.025	1.030	1.040	1.050	1.09	1.012	1.020	1.030
Heavy	1.035	1.050	1.030	1.040	1.045	1.060	1.10	1.015	1.025	1.035
In unheated premises at winter temperatures down to −20°C; low-temperature grease lubrication of bearings										
Moderate	1.040	1.050	1.035	1.040	1.045	1.065	1.10	1.025	1.035	1.045
Heavy	1.055	1.065	1.040	1.050	1.055	1.070	1.15	1.030	1.04	1.05

Table 8.7. Allowable Tensions (kN) of Heat-treated Hot-forged Detachable Chains for Moderate Operating Mode

Operating conditions	Chain speed, m/min	Chain link pitch, mm					
		80		100		160	
		Path configuration					
		simple	intricate	simple	intricate	simple	intricate
Light	up to 9	10	9	15	13	35	32
	9-18	9	8	13	12	32	30
	above 18	8	7	12	11	28	26
Moderate	up to 9	9	8	13	12	32	30
	9-18	8	7	12	11	28	26
	above 18	7	6	10	9	23	21
Heavy	up to 9	8	7	12	11	27	25
	9-18	7	6	10	9	23	21
	above 18	6	5	9	8	20	18

Notes. 1. The paths of simple configuration contain up to 20 horizontal and vertical bends, and the intricate ones, more than 20 bends.
2. For conveyers under light operating mode, the tensions indicated should be increased by 10% and for those operating under heavy conditions, decreased by 10%

or $600 \times 0.819 > 320 + 150$.

A graphical construction has shown that during the passage of loads on turning sprockets with a diameter $D_0 = 834.6$ mm there is enough space between adjacent load-carrying attachments, and therefore, their pitch has been chosen correctly.

To ensure the required throughput capacity, each attachment will carry one casting, i.e. $i = 1$. Then the speed of the conveyer will be

$$v = Za_s/(3600i) = 600 \times 0.6/(3600 \times 1) = 0.1 \text{ m/s}$$

We can adopt this speed, since it corresponds to the standard series of chain speeds. Loads will be supplied to the attachments by an automatic loading manipulator at a rate of one casting in 6 s.

Let us determine the linear gravity forces:

in the return run by formula (8.12):

$$q_0 = (m_b/a_s + m_t/t_c + m_{ch})\, g = (7.2/0.6 + 5.5/0.6 + 5.2) \times 9.81 = 260 \text{ N/m}$$

in the carrying run by formula (8.13)

$$q = q_0 + \frac{gm_l}{a_s} = 260 + \frac{9.81 \times 18.3}{0.6} = 560 \text{ N/m}$$

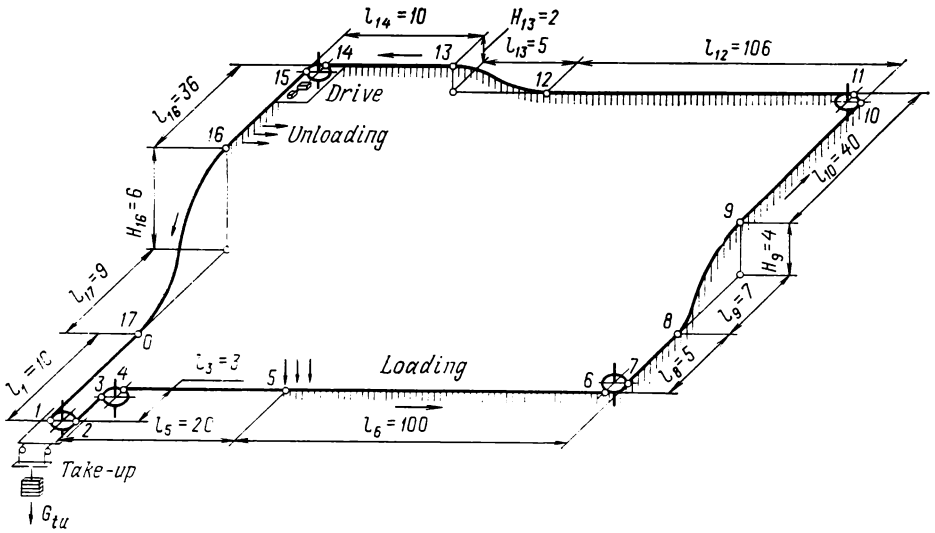


Fig. 8.29. Diagram for calculation of overhead load-carrying trolley conveyer (l and H in metres)

where it is taken: mass of attachment $m_a = 7.2$ kg; trolley pitch $t_c = a_a = 0.6$ m; trolley mass $m_t = 5.5$ kg; and linear mass of chain $m_{ch} = 5.2$ kg/m.

The highest chain tension is found approximately by formula (8.14):

$$S_{max} = S_0 K_l + w\,(qL_l + q_0 L_e) \times (1 + 0.3K_l) + qH$$
$$= 500 \times 1.2 + 0.027 \times (560 \times 309 + 260 \times 42) \times (1 + 0.3 \times 1.2) + 560 \times 6 = 10\,600 \text{ N}$$

where according to the conveyer scheme:

$$L_l = l_6 + l_8 + l_9 + l_{10} + l_{12} + l_{13} + l_{14} + l_{16} = 100 + 5 + 7 + 40 + 106 + 5 + 10 + 36 = 309 \text{ m}$$

$$L_e = l_{17} + l_1 + l_3 + l_5 = 9 + 10 + 3 + 20 = 42 \text{ m}$$

$$x = 6; \quad y = 4; \quad H = 6 \text{ m};$$

$$K_l = 1.02^6 + 1.025^4 \times 1.035 = 1.2$$

The resistance coefficients are found from Tables 8.5 and 8.6 for moderate operating conditions and load mass up to 50 kg: $w = 0.027$; $\varphi = 1.02$; $\xi_1 = 1.025$; and $\xi_2 = 1.035$. The highest chain tension obtained in the calculation, $S_{max} = 10\,600$ N, shows that the chain has been chosen correctly, since the allowable tension for it is equal to 13 000 N (see Table 8.7).

The drive is arranged at the highest point *14* of the carrying conveyer run (see Fig. 8.29) and the lowest chain tension should be expected after the descent at point *0* (*17*); this is the starting point for calculating the pull over the conveyer path [see formulae (8.15)-(8.18)]. We take that $S_0 = 500$ N. Then

$$S_1 = S_0 + wq_0 l_1 = 500 + 0.027 \times 260 \times 10 = 570 \text{ N}$$

$$S_2 = \xi_2 S_1 = 1.035 \times 570 = 590 \text{ N}$$

$$S_3 = S_2 + wq_0 l_3 = 590 + 0.027 \times 260 \times 3 = 611 \text{ N}$$

$$S_4 = \xi_1 S_3 = 1.025 \times 611 = 626 \text{ N}$$

$$S_5 = S_4 + wq_0l_5 = 626 + 0.027 \times 260 \times 20 = 772 \text{ N}$$

$$S_6 = S_5 + wql_6 = 772 + 0.027 \times 560 \times 100 = 2282 \text{ N}$$

$$S_7 = \xi_1 S_6 = 1.025 \times 2282 = 2340 \text{ N}$$

$$S_8 = S_7 + wql_8 = 2340 + 0.027 \times 560 \times 5 = 2420 \text{ N}$$

$$S_9 = \varphi(\varphi S_8 + wql_9 + qh_9) = 1.02 \times (1.02 \times 2420 + 0.027 \times 560 \times 7 + 560 \times 4) = 4920 \text{ N}$$

$$S_{10} = S_9 + wql_{10} = 4920 + 0.027 \times 560 \times 40 = 5530 \text{ N}$$

$$S_{11} = \xi_1 S_{10} = 1.025 \times 5530 = 5680 \text{ N}$$

$$S_{12} = S_{11} + wql_{12} = 5680 + 0.027 \times 560 \times 106 = 7280 \text{ N}$$

$$S_{13} = \varphi(\varphi S_{12} + wql_{13} + qh_{13}) = 1.02 \times (1.02 \times 7280 + 0.027 \times 560 \times 5 + 560 \times 2) = 8800 \text{ N}$$

$$S_{14} = S_{13} + wql_{14} = 8800 + 0.027 \times 560 \times 10 = 9000 \text{ N}$$

The highest tension has turned out to be at point *14*: $S_{14} = 9000$ N; it differs by 17% from that determined approximately: $S_{max} = 10\,600$ N.

In order to determine the tensions at point *16* and *15*, the calculation should be done against the direction of conveyer motion in the reverse order, beginning from point *0* (*17*):

$$S_{17} = S_0 = 500 \text{ N}$$

$$S_{16} = \frac{1}{\varphi}\left(\frac{1}{\varphi} S_{17} - wq_0l_{17} + q_0h_{16}\right) = \frac{1}{1.02}\left(\frac{1}{1.02} \times 500 - 0.027 \times 260 \times 9 + 260 \times 6\right) = 1940 \text{ N}$$

$$S_{15} = S_{16} - wql_{16} = 1940 - 0.027 \times 560 \times 36 = 1400 \text{ N}$$

The tractive force on the drive sprocket should be

$$W = S_{on} - S_{off} + W_{dr} = S_{14} - S_{15} + (\xi_1 - 1) \times (S_{14} + S_{15}) = 9000 - 1400 + 0.025 \times (9000 + 1400) = 8000 \text{ N}$$

The required power of the drive motor is found by formula (2.45):

$$N = \frac{Wv}{1000\eta} = \frac{8000 \times 0.1}{1000 \times 0.8} = 1 \text{ kW}$$

We select an electric motor of a power $N = 1.5$ kW and $n = 1000$ rpm.

The mass of the take-up weight will be

$$m_{tu} = \frac{1}{\eta_{pg}}(S_1 + S_2 + T) = \frac{1}{0.9 \times 9.81}(570 + 590 + 190) \approx 150 \text{kg}$$

8.2.4. Special Types of Overhead Conveyers

Very light goods (such as clothes or footwear) and small parts of a mass up to 5-10 kg are transported on overhead conveyers of a simplified design in which the tractive element is combined with the load-carrying element, or replaced by a rotating flexible shaft without traction. Conveyers of the first type (Fig. 8.30*a*) have a vertically or horizontally closed path. The tractive and the load-carrying elements are combined and made in the form of a synthetic fibre (nylon) cable or V-shaped rubber-fabric or rubber-cable belt which slides on continuous or periodically arranged steel or plastic supports. Load units are transported on hook attachments hanged directly onto the cable or belt. In conveyers with inclined sections of the path, the carrying belt has ridges and recesses on the top surface to prevent sliding of the load-carrying hooks (Fig. 8.30*b*).

Conveyers of the second type (Fig. 8.30*c*) have a rotating flexible

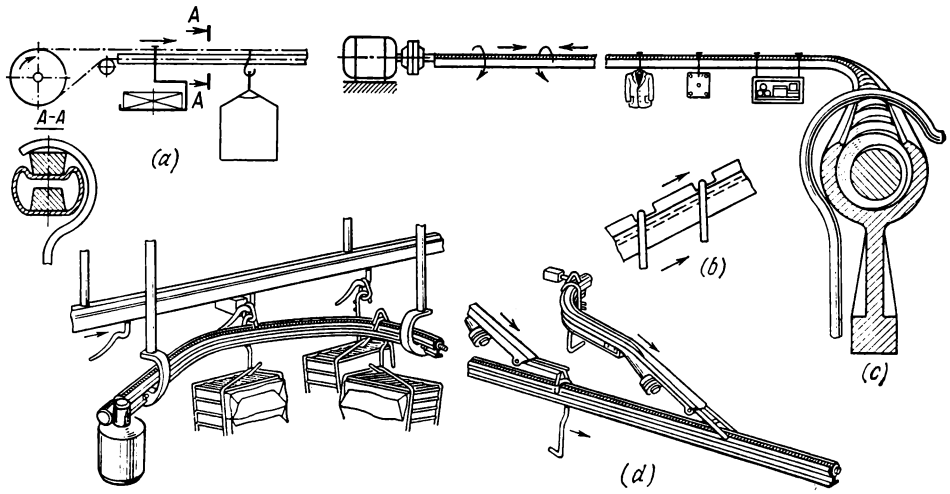


Fig. 8.30. Overhead conveyers for light loads
(a) and (b) with tractive and load-carrying elements combined; (c) with a rotating flexible shaft; (d) devices or load transfer from one conveyer to another

shaft—a steel cable (or sometimes a tube) with a diameter of 12.7-18 mm and steel wire wound helically onto its surface. The cable is driven by an electric motor and rotates in a semicircular groove of the supporting beam (or is supported by periodically spaced supports with plastic inserts). Hook attach ments with load units are hanged onto the cable and, as this rotates, move along the cable as a nut would move along a thread. Conveyers of this type have no return run, but can move loads in both directions (by reversing the cable rotation) on horizontal or inclined (up to 15-30°) paths and can transfer them to stores or other conveyers by means of tilting switches (Fig. 8.30*d*).

Special overhead conveyers have been designed for conveying long load units (timber, automobile chassis, prefabricated concrete panels of a length more than 6 m, etc.), which have traverse-type load-carrying attachments and guide wheels to ensure parallel motion of the loads on turning devices.

An interesting design has been proposed by combining a common overhead trolley conveyer and solid flight conveyer. Solid flights are fastened to the trolleys so as to move bulk loads in a trough suspended from the overhead conveyer track. Conveyers of this type may have a plane or spatial path with inclined and declined sections. The chain, trolleys and guide track arranged outside the trough are both isolated from the load and protected from its harmful effect. This method decreases the power consumption, increases substantially the service life and simplifies the conveyer assembly, maintenance and repairs.

An overhead flight conveyer can transport hot, toxic, and chemically aggressive bulk loads with the use of standard chains and wheels made of plain steel. An overhead belt conveyer is also known in which the trolleys carry cantilever attachments on which a troughed rubber-fabric belt runs along the entire length of the conveyer. Unloading is effected by straightening and tilting the belt on guide rollers. Loading can be performed at any point of the path onto the moving belt. This conveyer may have a horizontally closed or spatial path with turns in horizontal and vertical planes.

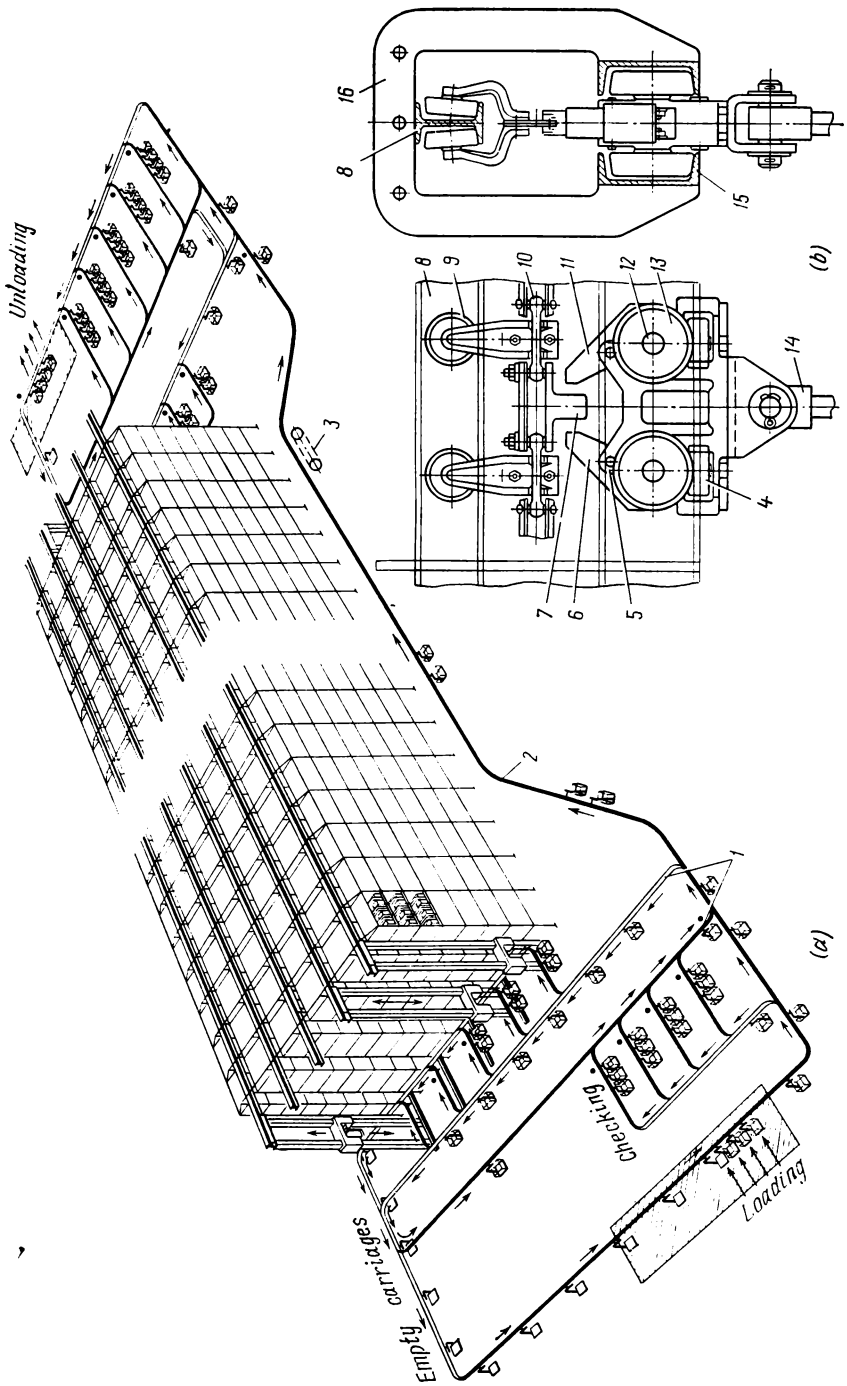


Fig. 8.31. An overhead pusher conveyer
(*a*) general view; (*b*) trolleys and pushers

8.3. Overhead Pusher Conveyers

8.3.1. General Design

An overhead pusher conveyer, called alternatively a power-and-free conveyer, has an endless hauling chain *10* with trolleys *9* fastened to it. The trolleys move on the upper (hauling) track *8*. The chain with trolleys is moved by a drive *3*. Pushers *7* fastened to the chain between trolleys or to trolleys proper move another group of trolleys *13* with load-carrying attachments *14* on another (carrying) track *15* which lies parallel to the first track in a vertical or horizontal plane. Both tracks are linked with brackets and fastened to ceiling structures of the building or separate supporting structures.

An overhead pusher (Fig. 8.31) conveyer differs from the overhead load-carrying type in that its chain is not fastened to the trolleys with load-carrying attachments and moves on a separate overhead track. Since the load-carrying trolleys are not fastened permanently to the hauling chain, but can be easily attached to and detached from it, and move on a separate track, they can be moved onto sideways by means of automatically controlled transfer devices or stopped on the moving conveyer in desired points of the conveyer path by means of automatic stops. The hauling chain is essentially of the same design as in overhead load-carrying trolley conveyers and is flexible horizontally and vertically, so that a pusher conveyer may have a spatial or horizontal plane path. Turns of the hauling chain in the horizontal plane are effected by turning devices or guide rails, and in the vertical plane, by vertical bends of the hauling and carrying tracks *2*. The carrying track of the conveyer may have sideways *1* to any side in the horizontal plane to transfer the trolleys onto another conveyer. Load-carrying trolleys can be transferred from one height to another by a vertical bend with inclined sections of the hauling and carrying tracks, or by means of a lowering section of the carrying track which can be moved upwards and downwards by a lifting mechanism. Load-lifting mechanisms are sometimes arranged on load-carrying attachments.

The hauling and carrying tracks of a conveyer may be arranged either one above another in the vertical plane (Figs. 8.31*b* and 8.32*a*) or in a horizontal plane (Fig. 8.32*c*). The pushing elements may be located either centrally above a track (Figs. 8.31) or sideways (Fig. 8.32*c*). With side pushers, the pushing force is applied eccentrically and creates a moment which tends to turn a trolley and chain in the plane of the joints. This moment may be rather high when the pushers have an appreciable length for side engagement of trolleys. In view of these drawbacks, side pushers are used rarely and are efficient only when the force applied to trolleys is low (up to 1.25 kN).

The wheels of load-carrying trolleys can move inside the track beams (see Fig. 8.31) or outside (Fig. 8.32*f*). Conveyers with the trolleys rolling above the track beams have an increased height in the cross section and can be recommended only for trolleys of a high load-carrying capacity (1250 kg or more).

Pusher conveyers with trolley wheels inside the beams may have a single-way track (Fig. 8.32*a* and *b*) or double-way track (Fig. 8.32*c-f*). In a single-way track developed in this country, two track beams (rolled or bent channels) serve simultaneously as the hauling and carrying track, i.e. the chain with the upper trolleys moves on the top flanges of beams and the load-carrying trolleys, on the internal surfaces of the bottom flanges. This design makes it possible to diminish the mass and dimensions of the moving parts and tracks of a conveyer, and therefore, to lower the cost.

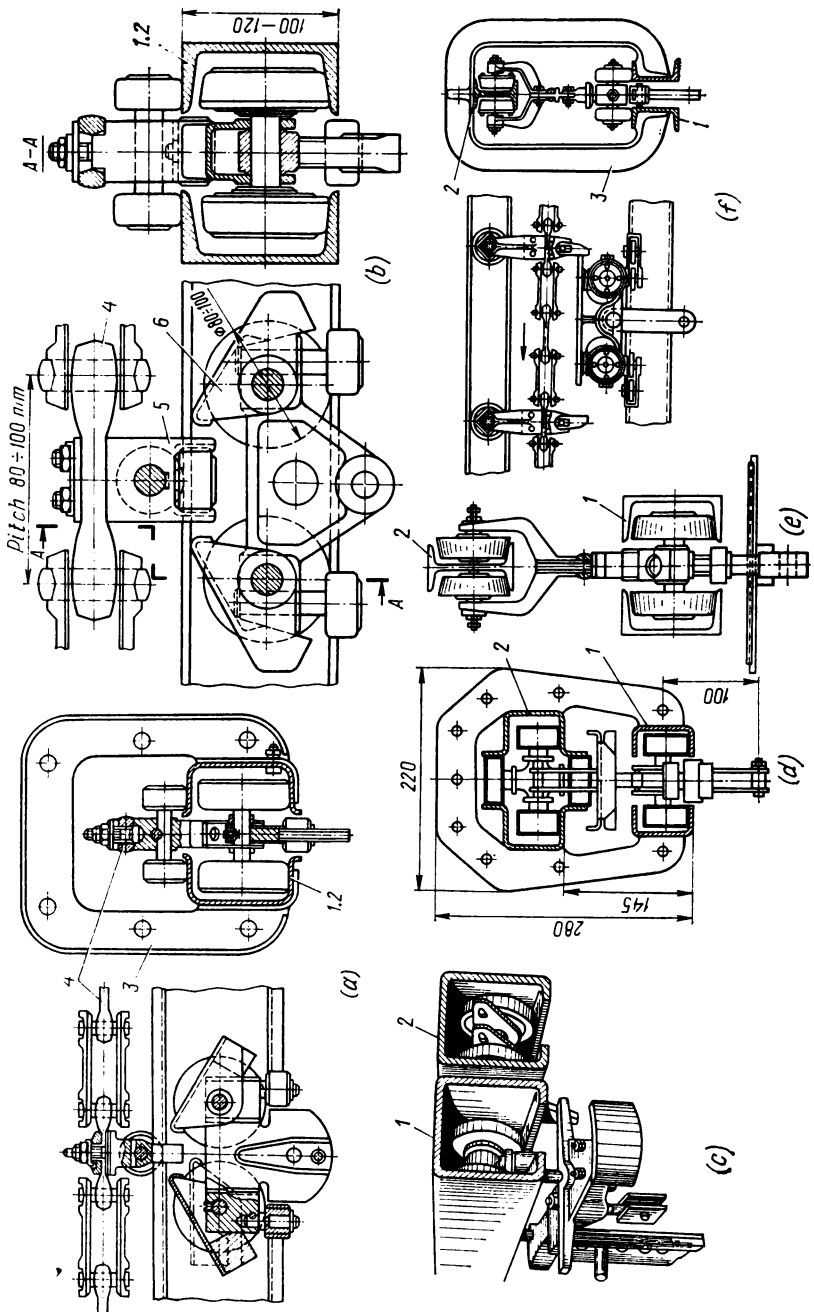


Fig. 8.32. Trolleys and tracks of overhead pusher conveyers
(*a*), (*b*) single track; (*c*) double horizontal track; (*d*)-(*f*) double vertical tracks; *1*—carrying track; *2*—hauling track; *3*—clamp; *4*—hauling chain; *5*—pusher; *6*—trolley stop

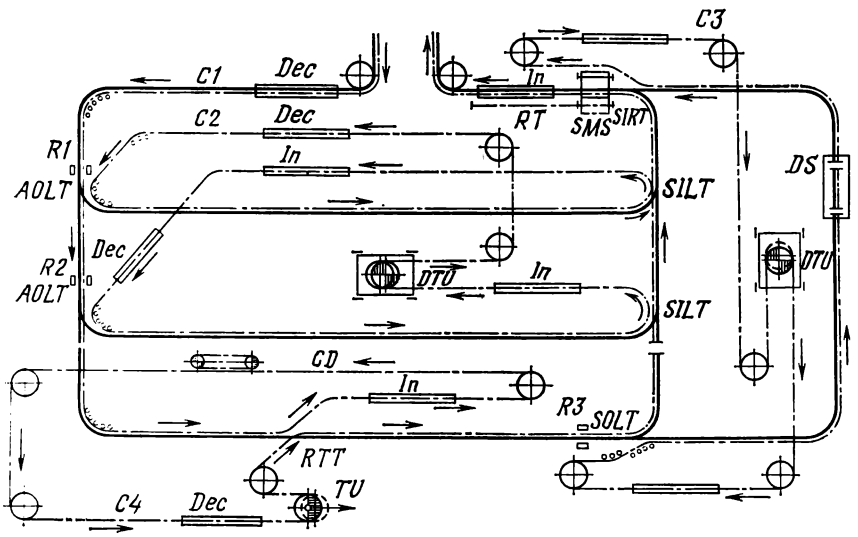


Fig. 8.33. Typical path layout for system of overhead pusher conveyers

C1, *C2*, *C3*, *C4*—conveyers; *R1*, *R2*—address readers; *DTU*—drive and take-up; *CD*—caterpillar drive; *SMS*—side-moving section; *DS*—lowering section; *RT*—repair track; *TU*—take-up; *In*—inc ine; *Dec*—decline; *AOLT*—angular opposite left-hand transfer; *SIRT*—straight inway right-hand transfer; *RTT*—right-hand through transfer; *AOLT*—angular inway left-hand transfer; *SOLT*—straight opposite left-hand transfer

In modern designs of pusher conveyers, load units (especially those having large dimensions and mass) are transported on traverses attached to a coupling of two or three trolleys provided with an automatic stop to disengage them from a pusher. Automatic stops widen substantially the possibilities for stopping and storing load units in the desired places along the conveyer path, but complicate noticeably the organization of and control over load flows.

Pusher conveyers can be employed effectively in complex automated conveying systems which combine various conveying and process lines and where the loads should be moved at different speeds. Such a system has a single load-carrying (distributing) conveyer path to move the whole set of load-carrying attachments, and numerous sideways which may have separate hauling paths. This makes it possible to avoid reloadings and ensure continuous transportation of loads through the entire technological process from the first to the last process operation. A typical layout of such a system with pusher conveyers (which comprises the main conveyer *C1* and auxiliary conveyers *C2* and *C3*) is illustrated in Fig. 8.33.

Automatic transfer of load-carrying trolleys with loads from the track of conveyer *C1* onto the tracks of the auxiliary conveyers *C2* and *C3* is performed by means of transfer devices with mechanical switches (in angular transfers) or without switches (in straight transfers). Since the transfers can move the loads onto auxiliary tracks, this makes it possible to perform process operations of different rate in the common transportation system, for instance, to separate and arrange various kinds of loads in a particular order for their assembly in an assembling line; to organize mobile intermediate stores where loads on load-carrying trolleys can be stored in one or more stages and delivered automatically in the required order to assembling lines; or to vary the spacings and speeds of load-carrying attachments without

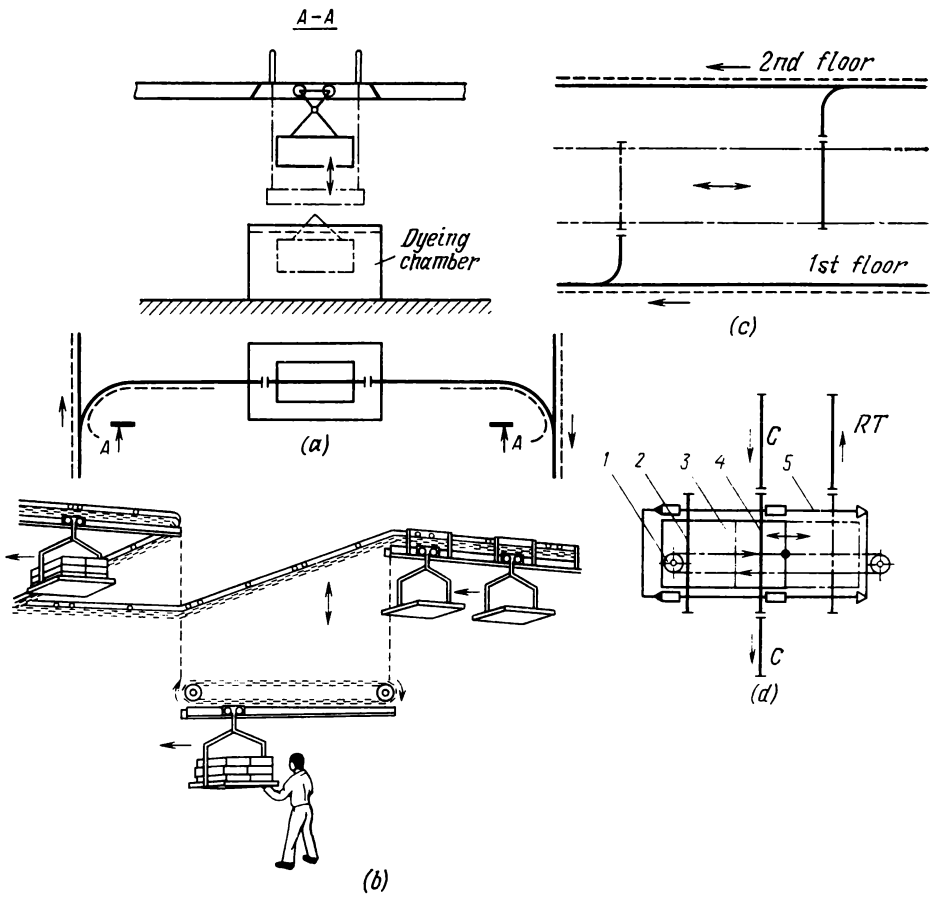


Fig. 8.34. Transfer sections of a pusher conveyer
(*a*) and (*b*) stationary lowering sections; (*c*) mobile lowering section (with an overhead hoist); (*d*) horizontal side-moving section; *1*—drive; *2* and *4*—sections of conveyer track; *3*—transfer carriage; *5*—carriage tracks; *C*—conveyer; *RT*—repair track

changing the rate of the production process.

Distribution of loads between sideways and transfer control are effected by means of automatic addressing: a load-carrying trolley has an address carrier set to a particular address; readers *R1*, *R2*, *R3*, . . . are arranged before the transfer means in the conveyer path and set up appropriately.

In a system of pusher conveyers, load-carrying trolleys can be transferred automatically to various levels by means of lowering sections of the carrying track which may be stationary (*DS* in Fig. 8.33, see also Fig. 8.34*a* and *b*) or movable by an overhead hoist (Fig. 8.34*c*). These devices ensure mechanized loading, unloading and transfer of loads and make it possible to place loads on working tables for assembly and perform various process operations (for instance to dip loads into a bath), or to transfer the load-carrying attachments from one conveyer onto another placed in the same plane or at a different level (Fig. 8.34*d*).

As loads are moved on a conveyer, they can be weighed automatically on weighting sections of the conveyer track, where their weight can be read off visually on an indicator or written

by an automatic recorder; it is also possible to count the load units moved on a conveyer by means of pulse counters, photo-cells or an automatic addressing system.

Modern pusher conveyers are complicated machines of a high class of accuracy. In addition to the conveyer proper they also contain a large number of auxiliary mechanisms, locking devices and automatic controls.

Overhead pusher conveyers are more intricate and expensive (1.5-2.5 times) than overhead load-carrying trolley conveyers. A pusher conveyer has a larger height of tracks and a greater (1.5-2.5 times) mass per unit length of the conveying path, since it has two tracks (hauling and carrying) and two groups of trolleys.

On the other hand, pusher conveyers, if employed rationally, are highly effective and can pay back in a relatively short time. They are employed in various branches of industry. Their application is economically most efficient in conveying process lines where various unit loads should be transported, processed and stored. Pusher conveyers are indispensable in automatic conveying systems which should combine different-speed transport and process operations, transfer the loads automatically from one line to another, and deliver supply units to assembly lines automatically (by a programme).

Pusher conveyers are not recommended for production lines in which the transport and process operations are performed at the same rate; in such cases overhead load-carrying trolley conveyers with automatic loading and unloading are preferable.

In the design of transport-process

Table 8.8. Main Characteristics of Overhead Pusher Conveyers (USSR)

Characteristic	Conveyer type				
	ТПВ-200Д	ТПВ-100	ТП-80	КТ-100	КТ-160
Maximum load-carrying capacity in horizontal sections, kg:					
single trolley	50	150	250	500	1250
two trolleys combined (traverse)	70	280	350	800	2000
Maximum angle of incline, deg.	60	30	45	30	30
Chain type	double-hinged	detachable		detachable PC100	special PC160
Chain pitch, mm	200	100	80	100	160
Rated tractive force of chain under moderate conditions, kN	4.5	12.5	8	12.5	30
Maximum speed of chain, m/min	40	30	27	18	18
Hauling track, I-beam No.		10	8	10*	10*
Carrying track, two channels No.	—	10	8	10*	15*,
Mass, kg of:					
head trolley	4.4	23	10	21	43
end trolley	2.4	21	7	14	35
straight track section 6 m long	88	221	158	205	345

* Special rolled sections.

Note. In conveyer type ТПВ-200Д, both tracks are made from bent channel profiles.

systems, it is essential to ensure independent interaction of individual parts of the complex. Then, should a failure occur in a separate part, it will not disturb the operation of the main line. This method increases the reliability of a system. Excessive saturation of a system with various interrelated mechanisms can make it too complicated and decrease substantially its reliability, since a failure of a single mechanism can stop the entire system.

The main characteristics of overhead pusher conveyers manufactured in this country are given in Table 8.8. The principal characteristic is the capacity of the load-carrying trolley which, in contrast to trolleys of load-carrying overhead trolley conveyers, is determined only by the mass of the load and attachment. The capacity of load-carrying trolleys may reach 1250 kg or, in a coupling of four trolleys on a traverse, up to 6.8 t. The speed of the hauling chain and trolley may be from fractions of a metre to 30 m/min. The total length of automatic conveying systems with pusher conveyers attains 8 km. In modern automobile works, the total length of pusher conveyers may be as high as 100 km, with the lengths of load-carrying trolley and pusher conveyers being in a proportion of 1:1 or 2:1.

A control system for pusher conveyers includes the same elements as in automatic load-carrying trolley conveyers; modern complex conveying systems with pusher conveyers are controlled by means of electronic computers.

8.3.2. Conveyer Elements

Many elements of pusher conveyers, such as hauling chains, trolleys, turning and take-up devices, safeguards, and drives, are essentially the same as in overhead load-carrying trolley conveyers. Only their specific elements will be described below.

Pushers. A pusher propels a trolley along the carrying track. It may have a rigid (Fig. 8.35*a*), or one-sided hinged fastening (Fig. 8.35*b*) to the hauling chain of trolley. With hinged fastening of pushers to the chain, two pushers are provided for a trolley: the leading (rear) pusher and the safe (retaining) pusher. The former moves a trolley on horizontal and inclined sections of the conveyer path, and the safe pusher retains it on declined sections. A one-sided hinge allows a pusher to tilt to one side only: for leading pushers,

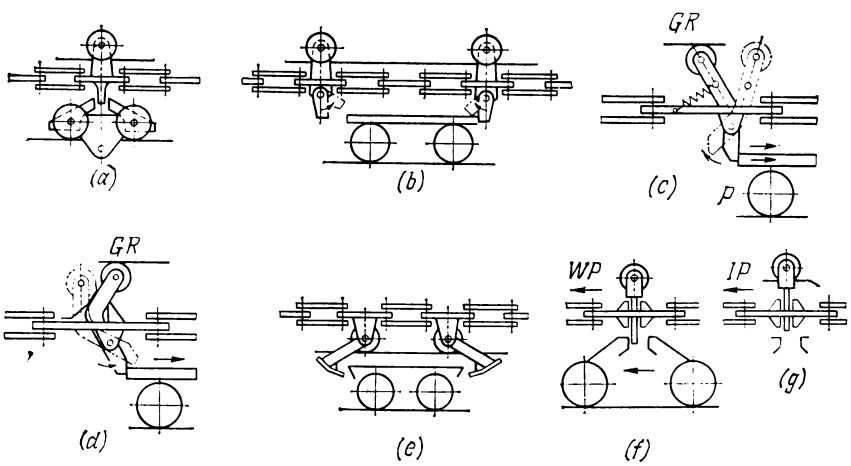


Fig. 8.35. Schematics of pushers
(*a*) and (*b*) uncontrollable; (*c*)-(*g*) controllable [(*d*), (*e*) tilting; (*f*), (*g*) extensible]; *WP*—working position *IP*—idle position; *GR*—counter-rail

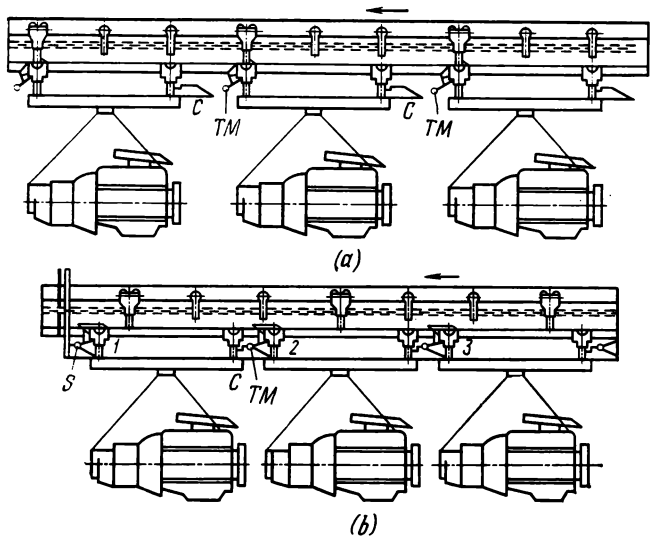


Fig. 8.36. Positions of load-carrying attachments with automatic stops (*a*) in motion; (*b*) in storage; *TM*—tripping mechanism; *C*—cutout; *S*—stop

towards the direction of chain motion and for retaining pushers, against that motion. With rigid fastening of pushers to the chain, each trolley is provided with two clamps with one-sided hinged fastening the trolley. One of them acts as a leading pusher and the other as the retaining. In both versions, a trolley can be easily moved into engagement with pushers and is retained from spontaneous movement.

In many cases, such as for performing various process operations or for storage, a trolley with a load must be stopped in a particular point on a horizontal section of conveyer path, i.e. must be disengaged from the moving chain. This is done by means of various pushers and clamps on the trolleys, which interact with a stop and thus disengage the trolley from the chain.

All pushers' may be divided into controllable and uncontrollable. The former may be tiltable in the plane of chain motion (Fig. 8.35*c-e*) or in a perpendicular plane or extensible (in a vertical plane), as in Fig. 8.35*f* and *g*. Controllable pushers interact with a guide bar and come into and out of engagement with a trolley without lifting or stopping the hauling chain. Uncontrollable pushers include rigid ones (Fig. 8.35*a*) and hinged pushers having a constant one-sided stop (Fig. 8.35*b*). They have a constant position relative to the chain and can be disengaged from a trolley only by lifting the chain section with the pusher or deflecting a hinged clamp on a trolley. Other versions of leading pushers are spring-biased and have an elastic spring-shaped one-sided stop (Fig. 8.35*c* and *d*), but they are used only seldom.

A particular design of pushers is chosen depending on the type and application of conveyer. All pushers must be strong and easily movable (without seizing in joints) into the working and idle position. Pushers and the chain links to which they are fastened are calculated for a bending moment which is the product of the maximum force to push a trolley with the heaviest load (this force is developed on the steepest portions of vertical

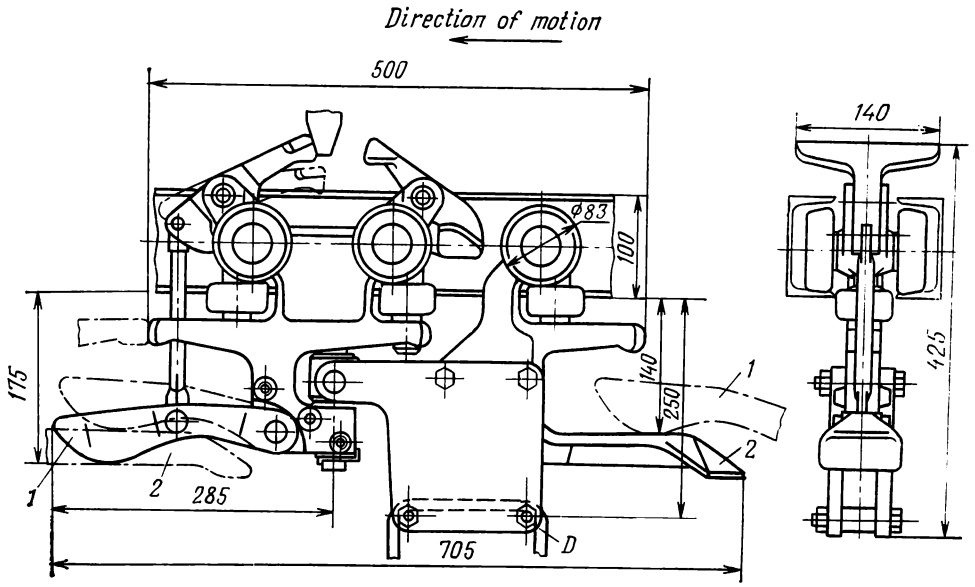


Fig. 8.37. A trolley with automatic stop

bends) by the arm of application of that force.

Automatic stops. An automatic stop is a mechanism which is mounted on a trolley and serves to disengage the trolley from a pusher of the hauling chain. Disengagement of conveyer trolleys from the hauling chain can be controlled by means of single or multiple stops.

A single stop can disengage only one trolley, those which follow behind that trolley will only be stopped if each of them has a mechanism for self-disengagement from the moving chain, an automatic stop. If a group of trolleys moving on the conveyer (Fig. 8.36*a*) must be stopped, for instance, for quality control, a stop *S* is switched on and stops the first trolley *1*. The chain continues to move and brings the next trolley *2* close to the stopped one (*1*), the disengaging mechanism (trip mechanism) *TM* of the trolley *2* comes into contact with the cutout *C* of the trolley *1*, interacts with the latter, and disengages the trolley *2* from the chain, so that it remains in place near the trolley *1*. The following trolleys (*3*, *4*, etc.) are stopped in a similar way (see Fig. 8.36*b*).

The mechanism of an automatic stop can be actuated only by a controllable pusher on the chain or controllable stop on a trolley.

It is distinguished between automatic stops with the lower (below the carrying track), upper (above the carrying track) and internal (inside that track) arrangement of the disengaging mechanism. The operating principle of an automatic stop with the lower arrangement of the mechanism (Fig. 8.37) is quite simple: a lug *1* on a moving trolley meets a step *2* on a previously stopped trolley and acts through a system of levers onto the trolley stop which is thus disengaged from a pusher of the hauling chain.

In an automatic stop with the upper arrangement of the mechanism, disengagement occurs by deflecting a controllable pusher or a stop on a trolley. Automatic stops should ensure steady position of the trolley and load-carrying attachment on disengagement from

the chain, have a small return stroke (recoil), and prevent oscillations of the load and spontaneous disengagement of the mechanism on a return movement of the trolley.

Trolleys. *Single trolleys.* A trolley moves a load-carrying attachment that is fastened to it along the carrying track. Trolleys may have two, four (Fig. 8.31), or six wheels; those on four wheels are the most popular. A four-wheeled trolley has a housing *5* (see Fig. 8.31) on which a load-carrying attachment *14* with a load is pivoted, four wheels *12* on two shafts, stops *6* and *11*, and guide rollers *4*. The guide rollers are needed to ensure the central position of the trolley on straight and curved portions of the conveyer path, and to actuate the switches. The shafts of wheels and rollers are fastened rigidly (see Fig. 8.31) or hingedly to the trolley housing.

Trolleys with rigidly fastened shafts are simpler in design, have a lower mass, are quite steady on straight portions of the track, but involve greater friction losses and require larger radii of horizontal turns. Both types (hinged and rigid trolleys) are employed in pusher conveyers, but those with rigidly fastened shafts of wheels are most popular. With rigid trolleys, the radius of horizontal turns of the conveyer path depends on the wheel base (the spacing between the centres of wheel shafts): a smaller wheel base allows a smaller turning radius.

Wheels of trolleys are mostly of flangeless type, made of steel with heat treatment of the rim to a hardness of 40-45 HRC. They are mounted on rolling bearings, as a rule on wheel-bearings of the same types and standard sizes as for trolleys of load-carrying overhead conveyers (see Fig. 8.5). Wheels of trolleys for a capacity ranging from 50 kg to 1250 kg have a diameter of 50-125 mm and wheel base varies from 100 to 180 mm.

The points of the pushing force application on a single trolley, and of the pivoted joint of a load-carrying attachment are spaced a certain distance apart, so that a moving trolley is acted upon by a moment which tends to turn it relative to the axis of the front wheels. In horizontal sections of the conveyer path, the pushing force on a trolley is not high, and the moment considered is inessential.

In inclined portions of the track, the force to push a trolley increases substantially, resulting in a corresponding increase of the moment. The latter depends on the design coefficient k which is the ratio of the distance b between the point of the pushing force application, and of joint of the load-carrying attachment to the wheel base a_1. The force developed by the trolley wheels on the track and a counter-rail is the sum of the normal component of the gravity force of the load and additional forces due to the moment considered.

In horizontal sections of a conveyer path, the pressures on the front and rear wheels of a trolley are equal to each other. With an increasing angle of incline of the track, the pressure on the front pair of wheels increases, and that on the rear pair, decreases up to zero, then changes the direction and increases again. The pressure on wheels increases sharply with an increase in the coefficient k. The design coefficient $k = b/a_1$ is an important factor which can influence the magnitude and distribution of pressures between the wheels; it is essential to minimize this coefficient in the conveyer design. In existing conveyers, $k = 0.526\text{-}1.3$. In circular portions of vertical bends, the trolley is not acted upon by the additional forces of chain tension, since it is not fastened to the chain. Wheel bearings of trolleys are calculated for the equivalent load (see Sec. 8.2).

At the same total load on a trolley, the pressures on the trolley wheels depend on trolley design (the coefficient k which is constant for a particular trolley design) and, with a constant

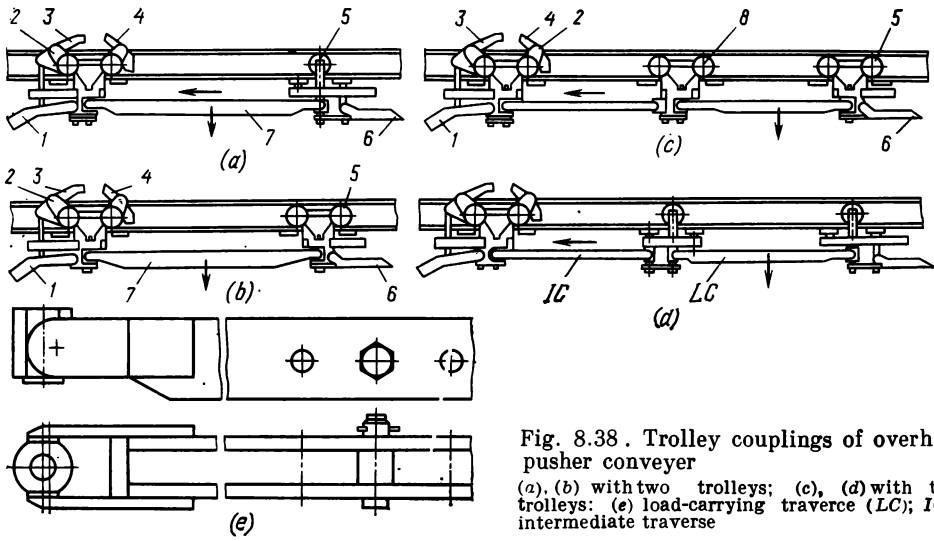


Fig. 8.38. Trolley couplings of overhead pusher conveyer
(*a*), (*b*) with two trolleys; (*c*), (*d*) with three trolleys: (*e*) load-carrying traverce (*LC*); *IC* — intermediate traverse

angle of incline of an ascending or descending track section, are independent of conveyer length or chain tension, i.e. are stable. (constant) for a particular design. For that reason, the load-carrying capacity of a trolley may be different in conveyers with a plane or spatial path. As distinct from load-carrying trolleys just discussed, the pressure on a hauling trolley depends on chain tension, which may vary in a conveyer depending on turns and bends of its path. The pressures on a trolley, as may be said, are individual parameters of each particular conveyer.

Multi-trolley couplings (traverse trolleys). Load units of a large size or mass are transported by a number of trolleys coupled by one or two traverses. A trolley coupling may have two (Fig. 8.38*a* and *b*) or three trolleys (Fig. 8.38*c* and *d*), and consists of a head trolley *2*, an end trolley *5*, an intermediate trolley *8* and doubled traverses *7* (Fig. 8.38). The head trolley has a controllable leading stop *3* and a safe stop *4* and an engaging mechanism *1* (trolley couplings with controllable pushers are also known). The end trolley *5* carries an end switch *6*. End and intermediate trolleys may be on two or four wheels depending on the total load on a coupling. Three-trolley couplings with an intermediate trolley are employed for loads of large overall dimensions and a large distance to their storage, where two-trolley couplings cannot be effective. In that connection, traverses are divided into load-carrying and intermediate (respectively *LC* and *IC* in Fig. 8.38). A load-carrying traverse is fastened to trolleys along their axes of symmetry and symmetrically to the supporting wheels, so as to prevent the appearance of a turning moment and skewing of wheels. The length of a coupling (usually 400-1000 mm) is limited by the condition that it must pass freely through conveyer turns and around turning devices. For that reason, it is advisable to have a long coupling made up of a number of short traverses.

The allowable load on a single trolley or coupling depends on the profile of conveyer path and the angles of incline.

Analysis of the allowable loads on a single trolley and multi-trolley couplings can show that the load does not increase in direct proportion to the

number of trolleys; this is an essential limitation for the application of couplings. The load-carrying capacity of a coupling is limited by the tractive capacity T of a pusher and the additional forces appearing in vertical bends.

In inclined portions (at an angle β) of the conveyer path, a pusher is acted upon by the longitudinal component of the gravity force of the load $m_l g$

$$F = gm_l (\sin \beta + w \cos \beta) \leqslant T$$

This force is absorbed only by the first leading trolley irrespective of the number of trolleys in a coupling. Since the strength of a pusher and of the chain link to which it is fastened is constant (it determines the tractive capacity of the pusher), the allowable mass of load will be lower at a higher lifting angle β. Therefore, the applicability of trolley couplings in spatial-path pusher conveyers is determined not by the number of trolleys in a coupling, but by the allowable strength of the pushers and chain and decreases at higher angles of the conveyer path incline.

Tracks. For light- and medium-duty pusher conveyers with trolleys of a load-carrying capacity up to 125 kg, the tracks are most often made from shaped bent sections of steel sheets 2.5-4 mm thick (Fig. 8.32*a*). Bent track sections can be manufactured by advanced stamping methods and have a relatively low mass per unit length, i.e. ensure a large saving of metal. It is not advisable to make conveyer tracks from four-six rolled angles, since their straightening and alignment requires much hand labour.

The tracks for medium- and heavy-duty conveyers with the trolley capacity above 125 kg are mainly manufactured from rolled sections: hauling tracks from I-beams Nos. 8-10 and carrying tracks, from two channels Nos. 8 and 16 (see Fig. 8.31). In conveyers with horizontally arranged tracks and side pushers, the tracks are made from tubes with a longitudinal cut (for light-duty operation), bent channel sections (medium-duty) or I-beams (for heavy-duty operation).

Conveyer tracks for increased service life are made of steel 09Г2. Track portions in vertical bends and horizontal turns, which are subject to intensive wear, are recommended to be heat-treated.

The path of a conveyer has straight sections, horizontal turns and vertical bends which consist of the inlet and outlet curved portions and straight portions of the slopes. The radii of turns and bends are chosen according to the type of chain and pitch of trolleys as recommended in Sec. 8.2.

The hauling and carrying tracks, the beams of a carrying track and counter-rails are connected together by clamps *16* (see Fig. 8.31) which are spaced at intervals of 1-1.2 m; additional clamps are provided at joints between track sections. The clamps maintain a constant gauge of tracks, i.e. hold them parallel to each other and increase their stiffness and steadiness.

In conveyers with the carrying track of a closed profile, i.e. with the trolley moving inside, say, two channels, inspection view-holes should be provided in the track beams and some portions of the track should be left open for trolley repairs.

The ends of track sections are joined by bolting to end clamps or, quite rarely, by welding. Extensible joints of tracks at the take-up and those for compensation of temperature expansions are made in the same way as in load-carrying trolley conveyers.

Conveyer tracks are attached by the clamps to structural elements of a building (trusses, ceiling beams, or columns) or to special supporting structures by means of ties or rigid struts essentially in the same manner as in load-carrying trolley conveyers.

During the trolley motion, the hauling and carrying tracks of a conveyer are subjected to total lateral bending,

local bending of beam flanges on the trolley wheels, and constrained (bending) torsion due to an eccentric position of trolley wheels relative to the vertical axis of track cross section which passes through the centre of bending. Track beams are calculated for bending strength and sag; the latter must not exceed 1/500 of the beam span. A greater sag may lead to excessive lateral oscillations of the tracks and disengagement of pushers from trolleys, especially at conveyer start-up. Further, a large sag increases the force to move the loaded trolleys. The total highest stress in a beam cross section is found as the sum of all component stresses and should not exceed 140 MPa for steel Ст. 3 or 160 MPa for steel 09Г2. Clamps of tracks are calculated as rigid frames for bending and tension under the action of a vertical and horizontal (compressive) load.

Transfer devices. Transfer devices, or simply transfers, are used in conveying systems for the transfer of trolleys or trolley couplings from one conveyer to the other, for instance, from a distributing conveyer onto an auxiliary one or vice versa. Their designs are diverse. A transfer device comprises a transfer mechanism, turning device, and a mechanical switch on the carrying track to direct the trolley from one track onto another. A method of direct transfer of trolleys from a pusher of one conveyer to a pusher of the other has greatly simplified the design of the transfer device and obviated the need in transfer mechanism (its function is performed by the hauling chain of one of the conveyers).

Transfer devices can be divided into straight and angular depending on the direction of transfer of trolleys from one conveyer onto the other. Right- and left-hand angular transfers are employed when the paths of two conveyers are at a certain angle (usually 90° or 60°) to each other. Straight transfers are employed when the two conveyers have parallel paths and the transferred trolleys continue to move on a straight track; examples are a right-hand through transfer *RTT*, straight opposite right-hand transfer or straight inway right-hand transfer SIRT (see Fig. 8.33). Inway transfers have inway tilting switches to combine two or more flows into a single flow (for instance, an angular inway left-hand transfer *AOLT* in Fig. 8.33). Transfer devices are called 'right-hand' or 'left-hand' depending on the position of the transfer and receiving tracks relative to a straight conveyer section, as seen in the direction of trolley motion. An opposite transfer device in which the trolleys are moved directly by pushers of a transfer conveyer *TC* and receiving conveyer *RC* (Fig. 8.39*a*) comprises a section of the main conveyer track *4*, an auxiliary track *1*, a roller set *7* to deflect the hauling chain of the auxiliary (receiving) conveyer, and a switch *3*. An opposite switch has track sections *4* and *1*, a switch tongue *6*, an axle *2* mounted in the housing, and a drive *5*. The tongue axle rotates freely in radial ball bearings mounted in the switch housing. The tongue is rigidly fastened to the axle and can occupy either the straight position (shown by dotted lines) or the sideway position (solid lines). The tongue is switched by means of the drive *5* (for instance, one or two electromagnets, pneumatic cylinder or annular linear asynchronous electric motor). The last type is advantageous since it allows up to 10^7 switchings. The drive of a switch is energized by a pulse sent from a reader of an automatic system of trolley addressing.

An inway switch (Fig. 8.39*b*) has no drive and its tongue is switched by the guide roller *8* of an approaching trolley. The roller should be adjusted so as to switch the tongue before the front wheel of the trolley runs on it. A transfer device without the transfer mechanism, in which the trolleys are moved directly by pushers of the hauling chains (by the 'pusher-to-pusher'

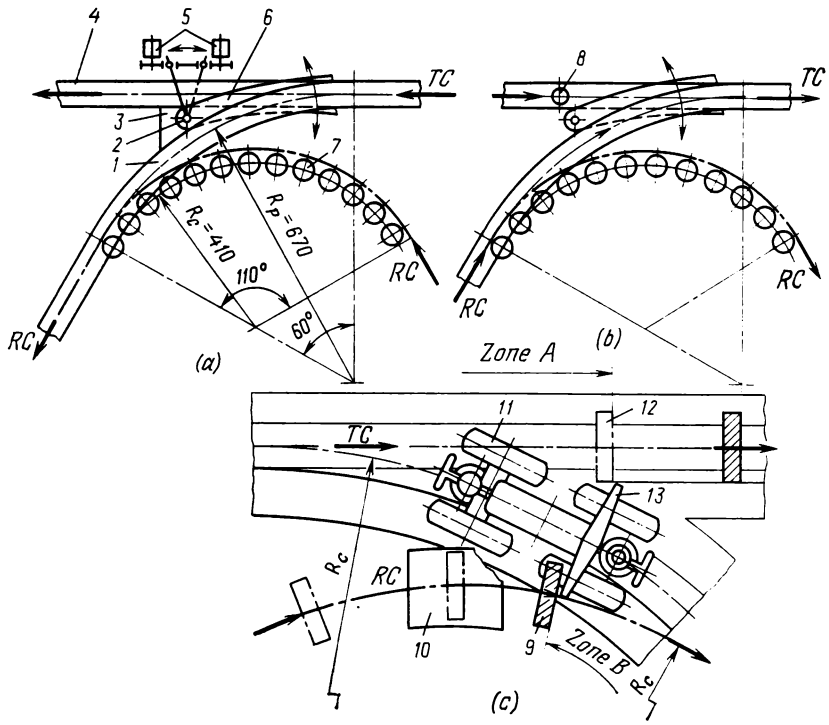


Fig. 8.39. Schemes of transfer devices
(*a*) opposite left-hand; (*b*) inway right-hand; (c) from pusher to pusher; *TC*–transfer conveyer; *RC*–receiving conveyer

principle) is illustrated in Fig. 8.39*c*. In zone *A*, a pusher *12* moves a trolley *11* onto a turn of the track. The trolley is then moved in zone *B* and along the path of the sideway (receiving) conveyer by a pusher *9* of the chain of that conveyer. A guide rail *10* is provided at the outlet of zone *A* (before zone *B*) to prevent the pusher *9* of the receiving conveyer from running into the side of a trolley which may be moved at that moment by pusher *12* on the track turn. The guide rail lowers (by 16 mm) the leading stop of the trolley (in a system with controllable pushers, a leading pusher is lowered), so that a pusher of the chain of the receiving conveyer can pass freely above the pushers of the trolley. On passing through the critical zone, the trolley stop comes out of contact with the guide rail and the trolley is catched by a next pusher of the chain of the receiving conveyer. In that connection, the hauling track of the transfer conveyer in the transfer zone is arranged closer to its carrying track (by 14 mm lower) and below the track of the receiving conveyer, so that the leading pushers of the transfer conveyer can be lowered and engage the leading stop of a trolley.

A coupling of a number of trolleys can be moved in track turns by engagement on the leading stop of the front trolley, with or without assisting pushing of the stop of the rear trolley. The latter method is more efficient, but requires a greater width of the pusher and the stop of the front trolley. Another drawback is that the rear trolley in a long coupling is subjected to a high lateral force in horizontal turns of the conveyer path.

Stops. A stop (single or multiple) is a mechanism for stopping a trolley in

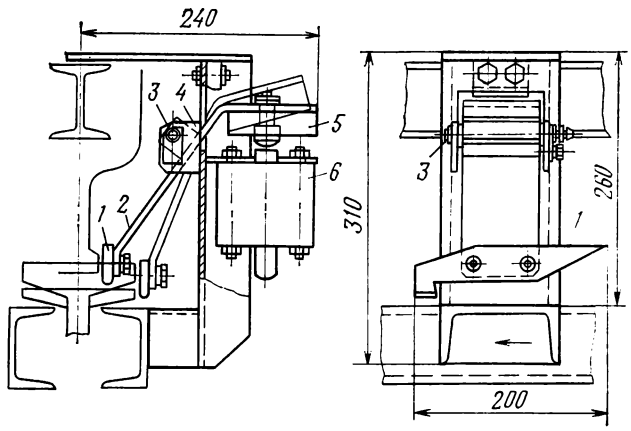


Fig. 8.40. An automatic stop for pusher conveyers

desired points of the carrying track of a conveyer.

Multiple (group) stops are mounted on storing tracks and comprise a head, intermediate and end stops which operate successively one after another. They are employed in conveyers where the trolleys are not provided with automatic stop mechanisms. Single stops are used in conveyers with the trolleys having an automatic stop which ensures stopping of a group of trolleys.

A single stop for trolleys with a wide leading stop (Fig. 8.40) has a rocker *2* pivoting on an axle *3* which is fastened in frame *4*. One end of the rocker carries a stopping plate *1* which can engage the stop of a trolley, and the other has a counterweight *5*. The rocker is normally held in the working upper position and is lowered only when the plate *1* comes into contact with the stop of a moving trolley; then it is disengaged from the pusher and the trolley is stopped. The stop is actuated from the control panel or a working place by means of an electromagnet *6* which attracts the counterweight *5* and thus moves the rocker out of engagement with the stop of trolley. In some designs, a pneumatic cylinder is used instead of an electromagnet.

Safeguards. Conveying systems with pusher conveyers employ essentially the same safeguards as load-carrying trolley conveyers (see Sec. 8.2), the only difference being that chain catchers in inclined conveyer sections are replaced by trolley arresters which have roughly the same operating principle as chain catchers and are mounted in similar places.

Lowering and side-moving sections. A lowering section is an arrangement for vertical (stationary section, see Fig. 8.34, *a* and *b*) or vertical and horizontal (mobile section, Fig. 8.34*c*) transfer of a portion of conveyer track with a trolley or trolley coupling from one level of the conveyer path onto another level.

A lowering section can move the load directly to working places or process operations (see Fig. 8.34*a*) or transfer the trolleys from the carrying track of one conveyer onto that of another, which may be at different levels in a vertical plane (one under the other) or in two vertical planes spaced a certain distance apart. A trolley is moved onto a lowering section by the hauling chain of the conveyer or by a special mechanism; the latter transfers the trolley onto the track of another conveyer.

Lowering sections exist in various types and designs. A lowering section usually has a mobile portion which can

move freely or in rigid guide beams, and a drive mounted on a fixed frame suspended from the building structures. The mobile portion comprises a section of the carrying track of the conveyer with a transfer mechanism (if any) and a rigid structure for connection to a hoisting element which may be in the form of two or four steel cables or two chains. The drive has a crane-type electric motor (usually with two speeds), brake, reducer gear, and sheaves for steel cables or sprockets for hoisting chains.

Typical lowering sections usually have the following characteristics: load-carrying capacity from 0.2 to 0.5 t; lifting height from 1 to 5 m; section length from 0.6 to 2 m depending on the overall dimensions of the loads and trolley couplings; working speed from 10 to 20 m/min; and the speed for placing loads, from 5 to 8 m/min.

A side-moving section (see Fig. 8.33, reference symbol *SMS*, and Fig. 8.34*d*) is used to move loads transversely on the same level in the horizontal plane to working places and for removal of trolleys and trolley couplings from the main conveyer track *C* onto an auxiliary repair track *RT*.

Automatic traffic control means. Reliable self-controlled operation of a pusher conveyer is ensured by various sensors of the automatic control system which are arranged at various points of the conveyer path. They ensure controlled, coordinated and successive operation of various elements of a conveyer and prevent the appearance of emergency situations. The moving chain with pushers, a trolley or a load-carrying attachment acts, in one or another way (mostly by mechanical contact), onto the probe of a sensor. The probe transmits this pulse to the sensor proper (for instance, a contact-type or contactless limit switch) which is connected into the control circuit of the conveyer. The sensor sends a signal to the control panel which sends a command to a particular actuating mechanism. The transmission of signals and commands may be electrical, pneumatic (for explosion-hazardous media), hydraulic or combined. Automatic control means control the movements of trolleys to and from free places, prevent collisions of trolleys, regulate the positions of trolleys on conveyer tracks, and perform a number of other control operations on a conveyer.

Though the sensors of pusher conveyers perform very diverse functions, they exist principally in three standard types: the trolley sensor, the pusher sensor, and the free pusher sensor.

In modern automatic control systems of pusher conveyers, the main type is an induction sensor in combination with a contactless switch and a corresponding relay. It is made in two embodiments: normally open and normally closed, which differ from each other in the form of screen for controlling the switch.

A pusher sensor (Fig. 8.41) comprises a lever probe *1*, an induction sensor *3* with a shaft *2* and a socket *4*, a bracket *5* and a clamp *6*; the sensor is fastened on the hauling track of conveyer and can interact with a pusher *T* of the conveyer chain. As the pusher of a moving trolley contacts the sensor probe, it deflects it by 30-50°. The probe turns the shaft *2* and the screen in the sensor body, thus actuating the sensor which sends a signal into the control system.

The principal schematics of arrangement of sensors on inway and opposite angular transfer devices are illustrated in Fig. 8.41*b*. A trolley on conveyer *C1* to be transferred onto conveyer *C2* is stopped by an arrester *S* (waiting position) and waits until a free pusher of conveyer *C1* comes into contact with it. As the free pusher contacts a free pusher sensor *FPS*, the latter sends a signal to de-energize the arrester *S* and to free the trolley to move onto an inway switch; the approaching free pusher takes the trolley and moves it along the path of conveyer *C1*. Trolley sensors *TS1* and *TS2* interlock the

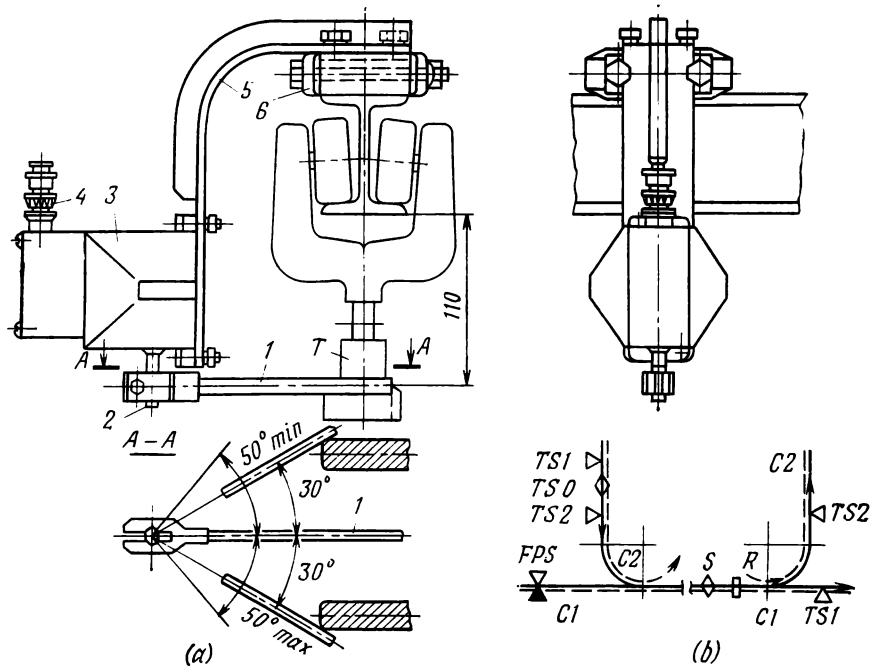


Fig. 8.41. Pusher sensor
(*a*) design; (*b*) schematic of arrangement in conveyer path

circuit so that the trolley can move safely without colliding with a next approaching trolley.

On approaching an opposite transfer device (Fig. 8.41*b*), the address carrier of a moving trolley interacts with a reader *R* and, if the latter is set to the required address, sends a signal to move the switch for entry onto conveyer *C2*; at the same time, a trolley arrester is switched over into the 'stop' position to arrest the next approaching trolley. The trolley sensors *TS1* and *TS2* de-energize the arrester after the trolley has passed through the switch and allow the passage of the next trolley; the sensor *TS2* then energizes the switch drive to move the switch tongue into the original straight position.

Automatic addressing system. The general principles of automatic addressing have been discussed in Sec. 8.2.

In pusher conveyers, the automatic addressing means should ensure automatic loading and unloading of trolleys and their motion along predetermined routes in the total system of conveyer tracks and their distribution through sideways between the addressed destination points.

An automatic addressing system and its characteristics are determined by the transportation-process route in which the load-carrying trolleys should pass through particular process operations. The system of address distribution (which includes switches and loading and unloading stations) in the path of a conveying system may be successive or parallel. In the former method, which is used at a low number of addresses (10-20), the addressed destination points are arranged successively along the conveyer path and their number is equal to the number of possible addresses in the address carrier of the system. In the latter method, group addressing is employed which allows a large number of probable addresses at a low capacity of the address carrier.

Modern overhead pusher conveyers mostly employ a decentralized magnetic system of automatic addressing without mechanical contact of information elements in the readers. Other systems are also known, for instance, with an electronic model on the central panel, with no information elements on conveyer trolleys, etc.

8.3.3. Conveyer Calculation

The calculation of a pusher conveyer requires essentially the same initial data as for an overhead load-carrying trolley conveyer. The minimal pitch of pushers (and therefore, the pitch of load-carrying trolleys) is determined, as for an overhead load-carrying trolley conveyer, by formula (8.11) to satisfy the condition that the largest load will pass freely on the steepest inclined sections of the conveyer path; it is taken that $\Delta = 0.2\text{-}0.3$ m in order to ensure wider gaps. Besides, it should be observed that the loads will pass unobstructedly on switches onto sideways.

The pitch of trolleys is determined by the radius of vertical bends (see Table 8.2) and the pitch of pushers and should be equal to four-six chain pitches.

The speed of the chain in the main conveyer path is usually taken within 0.3-24 m/min (see Table 8.4); the speed of auxiliary conveyers in sideways and the speed of transfer mechanisms of switches may be higher (but not more than 25 m/min) so as to ensure quick removal of trolleys from the main path. The pull of a pusher conveyer is calculated as for any conveyer with tractive element (see Ch. 2 and Sec. 8.2). The initial tension in the running-off section of the chain is taken at an elevated value ($S_0 = 1500\text{-}3000$ N) in order to prevent excessive deflection of pushers in the pushing plane and their spontaneous disengagement from trolleys.

The rated linear gravity forces (N/m) are determined separately for the hauling chain (q_0) and for the carrying chain in the unloaded and loaded state (q_1 and q):

$$q_0 = q_{ch} + g\,(m_t/t_c + m_p/a_s) \quad (8.19)$$

$$q_1 = g\,(m_t + m_a)/a_s \quad (8.20)$$

$$q = q_1 + g m_l/a_s \quad (8.21)$$

where all terms are as given earlier (see Sec. 8.2) and additionally m_p and m_t are the masses of pushers and trolleys, kg.

The highest tension of the conveyer chain is calculated by the generalized formula

$$S_{max} = S_0 K_l + [w q_0 L + w_1 (q L_l + q_1 L_e)]\,(1 + A K_l) + q H \quad (8.22)$$

where w and w_1 are the coefficients of resistance for hauling and carrying trolleys (see Tables 8.5 and 8.9).

Table 8.9. Resistance Coefficients w_1 of Trolleys on Wheel-bearings for Operation in Heated Rooms

Conditions of conveyer operation	Mass of load carried by a trolley, kg			
	up to 50	100	150	200 and more
Light	0.025	0.020	0.016	0.015
Moderate	0.032	0.027	0.022	0.020
Heavy	0.045	0.036	0.027	0.022

Detailed calculation of a conveyer is made by successive summation of the resistances in its sections, which can be found by the formulae:

for straight horizontal sections

$$S_n = S_{n-1} + (w q_0 + w_1 q')\, l \quad (8.23)$$

for horizontal turns on a sprocket or roller set

$$S_n = \xi S_{n-1} + \Phi i P_t \quad (8.24)$$

and for vertical bends

$$S_n = \varphi\,[\varphi S_{n-1} + (wq_0 + w_1 q')\,l \pm (q_0 + q')h \qquad (8.25)$$

where q' is the linear gravity force in the carrying run, N/m ($q' = q$ in its loaded sections and $q' = q_1$ in empty sections); Φ is a generalized coefficient of resistance for a single trolley in horizontal turns; i is the number of trolleys in a turn; and P_t is the load on a trolley including the gravity force of the trolley and load-carrying attachment, N.

The resistance coefficients ξ and φ have been given in Table 8.6. The coefficient Φ of resistance of rigid-base trolleys in horizontal turns is taken depending on the angle of turn and operating conditions of a conveyer

	90°	180°
light conditions	0.05	0.08
moderate	0.06	0.10
heavy	0.07	0.12

Multi-trolley couplings are subjected in curved portions of conveyer path to additional stresses, since a coupling occupies positions along chords of the turning arc.

8.4. Overhead Push-carrying Conveyers

The type of overhead push-carrying conveyer, which has been first developed in this country, is a combination of load-carrying and pusher trolley conveyer; it operates as the former in some continuously moving portions of the conveyer path, say, in the main line, and as the latter in other portions where the loads are delivered for storage, distributed, or stopped for performing process operations which are impossible on a moving conveyer.

An overhead push-carrying conveyer with trolleys of a load-carrying capacity of 500 kg has a hauling track *1*, trolleys *2*, a chain *3* (Fig. 8.42*a*), drive, and turning and take-up devices of the same design as in a load-carrying trolley conveyer.

A trolley has a pivoted hook-shaped pusher *5* with thrust rollers *6*. Another group of trolleys (trucks) *8* with load-carrying attachments have a bracket *7* at the top to engage the hooks of the upper trolleys. In load-carrying portions of the conveyer path, the trucks are carried by the trolleys in suspended state essentially in the same way as is done in a load-carrying trolley conveyer. The sections of the path where process operations should be performed have carrying tracks *9* with guides *4* for the rollers of hook-shaped pushers. As a truck approaches such a section, it enters the guiding mouth of the carrying track, is lifted and disengaged from the trolley hook, and further pushed by the top portion of the hook-shaped pusher as in an overhead pusher conveyer.

The conveyer sections with carrying tracks may have various sideways with switches, transfer devices and lowering sections as in a pusher conveyer. The trucks are distributed between sideways of the load-carrying track by an automatic addressing system. At the end of a carrying track, a trolley with a hooked pusher is raised along a vertical bend of the hauling track. The hook engages the bracket of a truck, lifts the latter above the carrying track and carries it further as in a load-carrying trolley conveyer. All switching operations on the conveyer are performed automatically.

A light-duty push-carrying conveyer (Fig. 8.42) has trucks *1* with a load-carrying capacity of 50 kg having a T-shaped bracket *2* and trolleys *3* with a roller grip *4*. The grip of a trolley is always under the side flanges of the bracket which excludes disengagement of the trolley and truck. This increases largely the safety of conveyer operation, since a truck cannot disengage from its trolley and fall onto the floor. In pushing sections of the conveyer path, trucks are moved by hinged

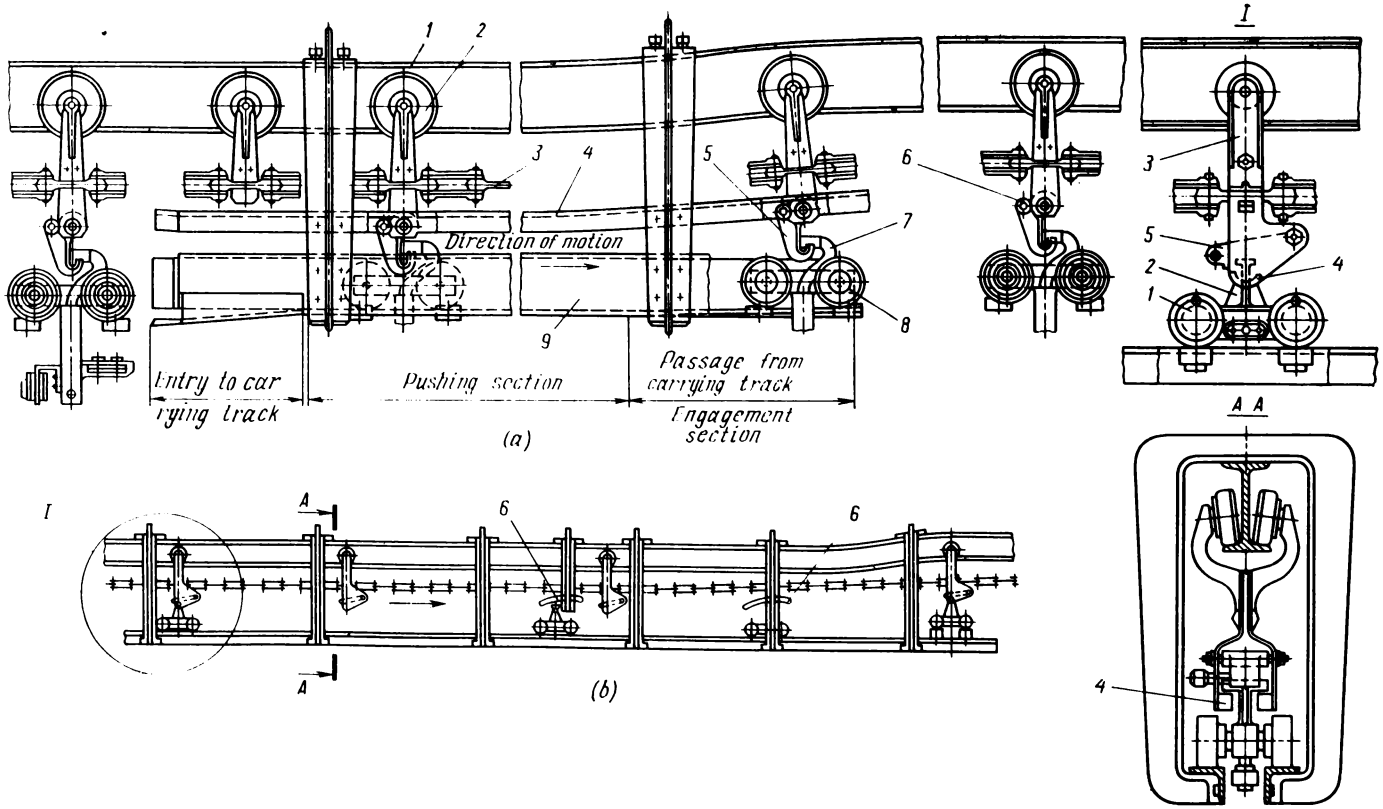


Fig. 8.42. Overhead push-carrying conveyers
(*a*) medium-duty; (*b*) light-duty type

pushers *5* and can be disengaged from them by an arresting rail *6*.

Push-carrying conveyers can be recommended for transporting of load units of a mass of 50-500 kg at relatively low speeds of the chain (roughly up to 12 m/min) in cases when the conveyers serve mainly for transportation and sideways for process operations have a relatively small total length (for instance, less than 40 % of the total length of the conveyer path) and also in conveyer lines between shops.

8.5. Overhead Load-towing (Floor-type) Truck Conveyers

In an overhead load-towing truck conveyer, load units to be transported are placed onto an on-floor carriage (truck) which can be moved on the floor of the building by means of a grip (see Fig. 8.1*c*) or a pusher (Fig. 8.43) mounted on an overhead trolley. The truck has a vertical mast to engage a grip or pusher of the trolley.

The mast of the truck may carry any type of address carrier of an automatic addressing system, which makes it possible to distribute the trucks automatically by means of switches provided along an overhead guide track for rollers of the mast. In sideways, the trucks are moved by trolleys of auxiliary overhead conveyers; their movement in on-floor car conveyers is also possible. By means of arresters, a truck can be disengaged from the moving chain in any point of the conveyer path.

Conveyers of this type offer the following advantages: the trucks can be easily engaged to and disengaged from the hauling chain without interrupting the continuous flow of goods along the conveyer path; they can transport heavy load units (of a mass of up to 2.5 t or more), since the weight of the load is transmitted onto the floor, rather than on the overhead track; they offer wide possibilities for automation and for interaction with on-floor electric truck loaders if load units are transported on bottom plates; they require no slits and recesses in the floor, as is the case,

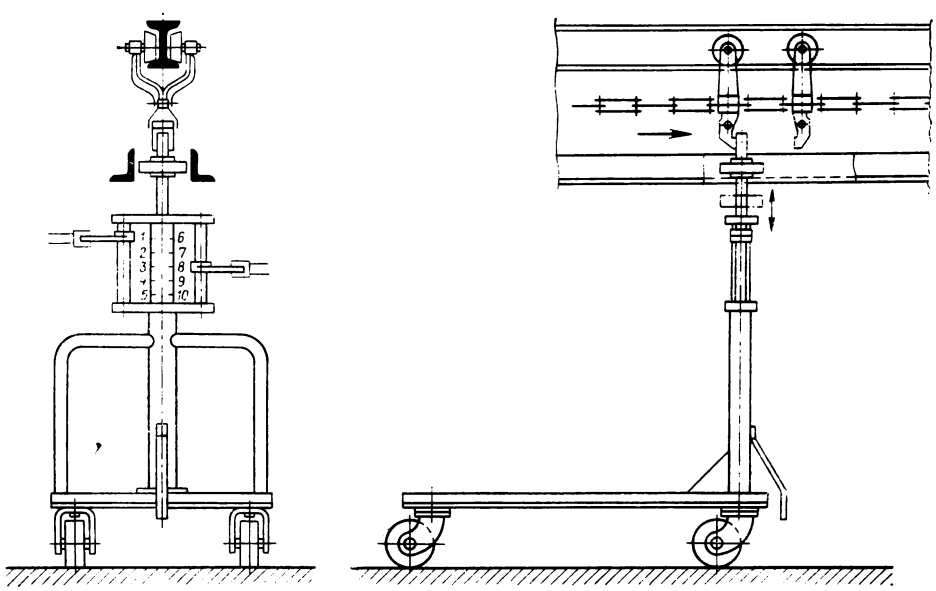


Fig. 8.43. An overhead load-towing truck conveyer with automatic addressing

for instance, with some types of on-floor car conveyers (see Ch. 9); all electrical and mechanical equipment is readily accessible for inspection; and finally, any changes in the conveyer path can be made rather easily. A drawback is that the tractive force is applied high above the floor which in some cases may result in overturning of trucks with heavy loads on inclined portions of the path.

Conveyers of this type have essentially the same characteristics as overhead load-carrying conveyers, but allow a higher speed (up to 45 m/min), since there is no risk of tilting of loads; on the other hand, they require larger radii of horizontal turns (3-5 m) and smoother inclines (with a slope up to 10-15%). Horizontal turns are made on roller sets.

Load-towing truck conveyers are employed in stores, luggage departments and other places where unit loads should be sorted and distributed over a large area.

The calculation of these conveyers is done as for pusher conveyers. The resistance coefficient of on-floor trucks depends on the type of the floor and the tyres of truck wheels (Table 8.10).

8.6. Overhead Tow-carrying Truck Conveyers

In some instances, it is essential to convey loads on trucks to various floors of a building without overloading. The problem has been solved by developing a combined tow-carrying truck conveyer which operates as a floor-type truck conveyer on horizontal portions of the path (on the floor) and as an overhead load-carrying type, on inclined and declined sections. In a simple conveyer design for light loads (up to 50 kg), a truck is attached to a trolley by means of a lever clamp (Fig. 8.44). On turning the clamp handle, a truck is attached to a trolley, lifted above the floor and transported along steep

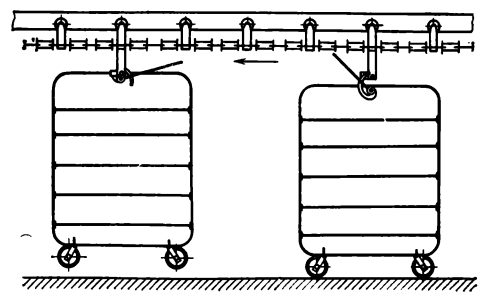


Fig. 8.44. Light-duty overhead tow-carrying conveyer with lifting levers on trucks

inclines and declines as a load-carrying attachment of an overhead trolley conveyer. When the handle is turned back in the desired place, the truck is lowered onto the floor and moved as in a tow conveyer.

A version of this type of conveyer is Giordani conveyer (Fig. 8.45) where the trucks have a hook *3* to engage a pin *1* arranged between catching plates *2* on a trolley of the conveyer. In an inclined section of the conveyer path (Fig. 8.45*b*), a truck is transported in the suspended state, with its hook *3* being in engagement with the trolley pin *1*. At the beginning of a horizontal path section, the truck is lowered onto the floor and further moved on it. In that case, the pin *1* (Fig. 8.45*c*) acts as a pusher by contacting the vertical surface of the hook *3*. The hook *3* has a

Table 8.10. Resistance Coefficients of On-floor Trucks on Rolling-bearing Wheels under Moderate Conditions

Type of floor	Truck tyres	
	solid rubber	pneumatic
Asphalt	0.033	0.025
Concrete	0.028	0.020
Wood	0.030	0.024
Compacted soil	—	0.080
Dry sand	—	0.150

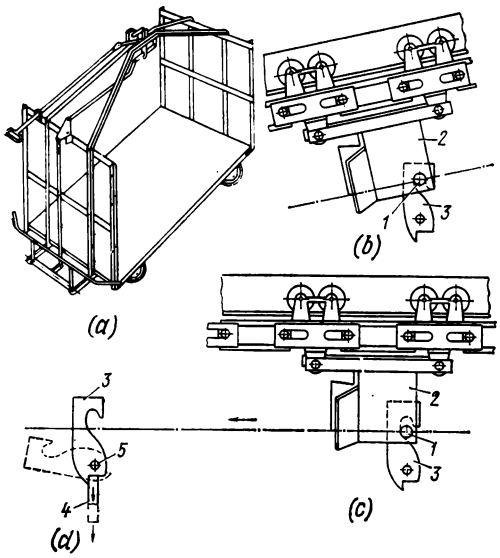
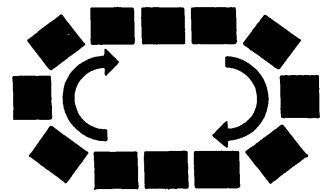


Fig. 8.45. Giordani conveyer (Italy)
(*a*) truck; (*b*) trolley with truck attachment; (*c*) truck-pushing section; (*d*) disengagement of truck hook from conveyer chain

stop *4* which prevents its tilting on axle *5* (Fig. 8.45*d*). The stop can be lowered by pressing a foot pedal, so that the hook is tilted and disengages from the trolley and the truck is stopped. Giordani conveyers are employed for conveying of load units of a mass of 125-1000 kg; they have essentially the same elements as load-carrying conveyers and the characteristics similar to those of tow conveyers, but a lower conveying speed (usually up to 16 m/min), which is required for safe engagement and disengagement of hooks on inclined sections.

9

Load-carrying Car Conveyers



9.1. Principal Types and Applications

A load-carrying car conveyer has a closed contour of endless hauling chain to which load-carrying cars (platforms) are permanently attached. The cars with loads move on guide tracks. Both the cars and the tracks are components of a conveyer. In this respect, they differ principally from load-towing car conveyers which will be discussed in Ch. 10.

Load-carrying car conveyers are employed for the transportation of loads between various operations in production process flow lines, for instance, for conveying parts and units from one working place to another in assembly lines, for moving casting moulds as they are assembled, filled, cooled, knocked out, and returned in a foundry shop, and in many other similar applications. In view of these numerous applications, load-carrying car conveyers are quite diverse in design.

The principal features for classification of load-carrying car conveyers are the arrangement of the tractive element and guide tracks and the position of cars on conveyer sections. By the first feature, they are divided into vertically closed and horizontally closed. The latter may either lie in a single horizontal plane or have a spatial path. As regards the position of cars on the return conveyer runs, the vertically closed conveyers may be with tipping or non-tipping cars. In horizontally closed conveyers, the cars may move on the floor or on tables (say, at the height of the tables of working places). As regards the kind of motion of loads, conveyers are known with a continuous (constant or controlled speed) or with intermittent (pulsating) motion of the cars.

The selection of a particular type of conveyer depends on the kind of technological process to be attended by a conveyer, the characteristics of the loads, and the layout of the production premises.

Vertically closed conveyers are more compact, since their return run moves under the carrying run. With tipping cars, however, only the upper run is utilized or the design of conveyer must be complicated substantially. With an appreciable time required for completing the production process, the length of a vertically closed conveyer becomes too large and may involve difficulties in the layout of the premises.

In horizontally closed conveyers, both runs are utilized efficiently, but the conveyer occupies more space. In conveyers of this type, the loads can circulate without being taken off from the cars, which makes it possible to utilize the conveyer of a relatively

short length in prolonged production processes (for instance, drying or cooling of articles on a conveyer) or as a moving store.

The kind of loads is also important for selection of a conveyer type. Heavy and bulky articles (such as automobiles, tractors, engines, etc. or their units) are, as a rule, transported on vertically closed conveyers, whereas horizontally closed conveyers are employed for light and small loads. Process conveyers for processes with prolonged and different in time operations (such as conveyers of foundry shops or test stands) are usually of the horizontally closed type with a plane or spatial path.

The principal characteristics of load-carrying car conveyers are the load-carrying capacity and the dimensions of car platforms, which in turn are determined by the overall dimensions and mass of the loads to be conveyed. It is advisable generally to place unit loads on car platforms with their longer sides along the conveyer axis and to choose accordingly the dimensions of cars (their length l and width B). It is usually taken that $l/B = 1.25\text{-}2$ and only in rare cases, $l = B$. With the recommended length-to-width ratios, it is possible to employ more efficient long-link hauling chains and to pass the loads more conveniently through process operations. In some cases, long narrow articles are laid across the length of cars or often onto two cars moving on parallel tracks (in double-chain conveyers). The standard series of car width is: 200, 320, 400, 500, 650, 800, and 1200 mm; the load-carrying capacity of conveyer cars may range from 10 to 8000 kg.

It may be taken conditionally that the group of vertically closed car conveyers includes carrier-chain conveyers in which the loads transported (such as automobile and tractor chassis) are placed onto special supports (seats), which are fastened rigidly to the plates of hauling chains, or directly onto widened chain plates. The role of conveyer cars is then taken directly by the hauling chains with running wheels.

A special group of car conveyers is represented by distributing table-type car conveyers with automatic unloading of cars at working places by an automatic addressing system. These conveyers may be vertically or horizontally closed or in some cases have a spatial path.

9.2. Vertically Closed Car Conveyers

9.2.1. Design

A vertically closed car conveyer with tipping cars (Figs. 9.1 and 9.2) has a tractive element *2* (see Fig. 9.1) with load-carrying cars *3* fastened to it. The cars move on guide tracks *4* of the conveyer structure by means of a drive *1* and take-up *5*. Assembly-line conveyers are usually provided with assembly rigs *1* (Fig. 9.2), turning devices *2* and other appliances which can facilitate assembling. The conveyer structure is protected at the sides by an enclosure *4* made from steel sheets or net and the upper conveyer run is separated from the lower one by a continuous steel-sheet trough *3*. The trough prevents injuries to assembly workers and falling of tools and parts onto the lower conveyer run and serves as an oil collector.

Conveyer cars are fastened rigidly to links of the hauling chain. The conveyer may have one or two chains, depending on the track gauge and the base of cars and the required tractive force. It may be taken approximately that with the track gauge A being up to 4 chain pitches, a single chain can ensure steady position of cars, where at A greater than 4 chain pitches two hauling chains should be preferably employed. The cars are guided during tipping over the drive sprocket by means of stationary counter-rails on

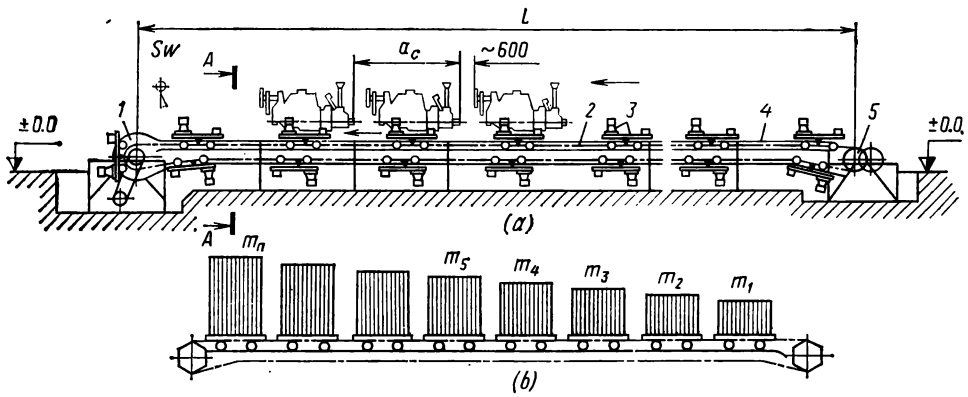


Fig. 9.1. Vertically closed car conveyer with tipping cars for assembly of automobile engines
(*a*) scheme; (*b*) diagram of loads on cars

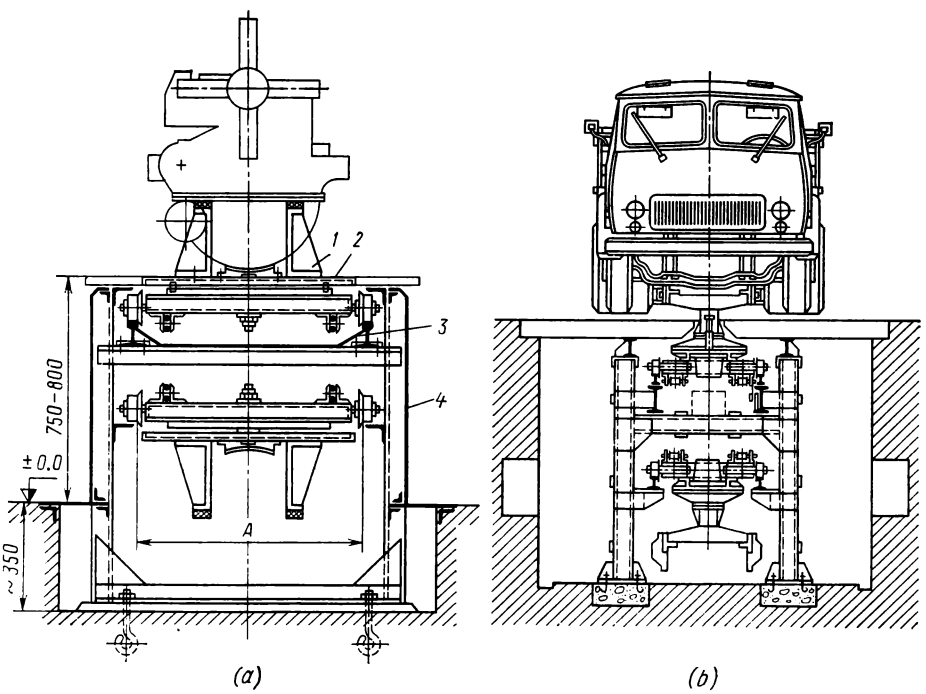


Fig. 9.2. Cross-sectional view of tipping-car conveyers
(*a*) for assembly of automobile engines (see Fig. 9.1); (*b*) for assembly of automobiles

which the car wheels run during car tipping, while the take-up sprocket has two guide discs set on its shaft (Fig. 9.3), which are moved together with the sprocket and shaft during chain tensioning. The diameter of the discs is chosen such that the frames of cars are pressed against them in the turns under the action of the chain tension.

Car conveyers with non-tipping cars (Fig. 9.4) have two hauling chains arranged at both sides of cars. Each car has two wheel axles. The driven axle *DA* is hingedly attached to the hauling chains and the idle axle *IA* rotates

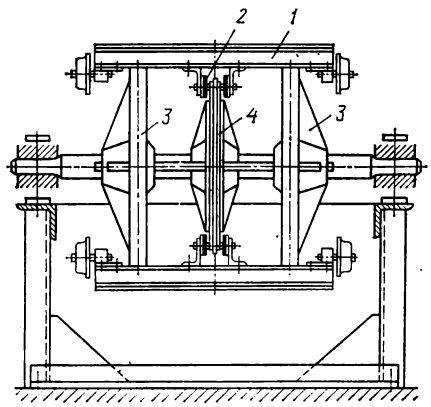


Fig. 9.3. Take-up sprocket with guide discs
1—car; *2*—hauling chain; *3*—disc; *4*—sprocket

freely. The drive and take-up sprockets are mounted on cantilever shafts to leave a free space in the middle for the cars to pass. In the turns of a conveyer path, the cars can move slightly tilted aside (Fig. 9.4*a*) or perform a plane-parallel motion and remain horizontal along the entire length of a turn (Fig. 9.4*b*). In both methods, the desired kind of car motion is ensured by means of guide tracks and mechanisms.

Conveyers with non-tipping cars can have only a straight horizontal path. The cantilever mounting of sprockets, an intricate fastening of cars to chains, and the necessity to provide additional mechanisms and guides in turn sections make these conveyers too complicated and expensive, because of which they have found no wide application. Their advantage is that loads can be moved both on the upper and lower run. This is made use of in foundry shops: casting moulds are assembled and filled with molten metal on the upper run, while cooled and delivered to the knock-out section, on the lower run. Conveyers of this type are employed in foundry shops for carrying relatively light moulds (up to 300 kg) and also in other cases when the production areas are restricted (especially the length), but the conveyer must have an increased length to move the loads through process operations of different rate (such as assembling, casting, cooling, painting, drying, etc.).

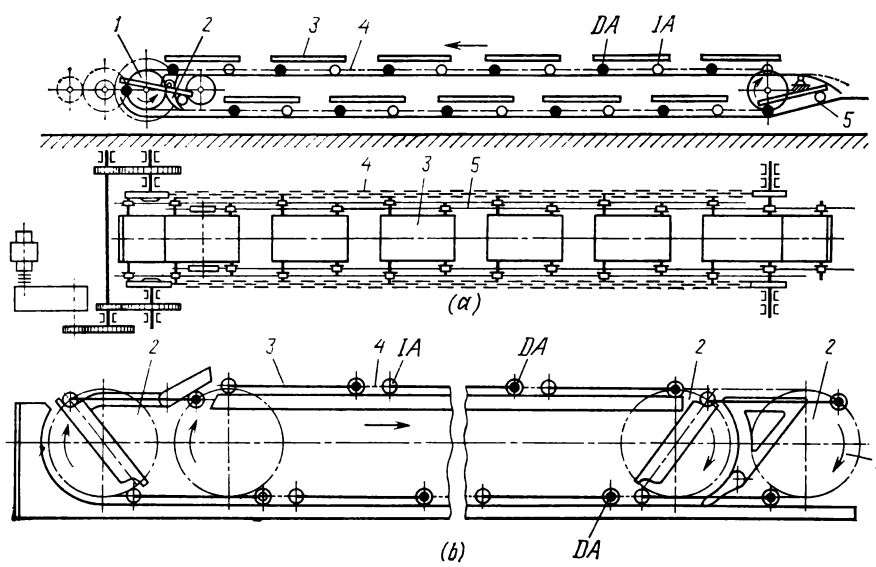


Fig. 9.4. Vertically closed non-tipping-car conveyers
(*a*) with cars inclined on turning devices; (*b*) with cars remaining plane-parallel; *1*—drive; *2*—arrangement for run-to-run transfer of cars; *3*—car; *4*—chain; *5*—track; *DA*—driven axle; *IA*—idle axle

9.2.2. Conveyer Elements

The tractive element of vertically closed car conveyers is made up of one or two chains: sleeve-type or roller laminated chains with a pitch of 80-320 mm (most often), detachable chains with a pitch of 100 mm or 160 mm, combined laminated chains or sometimes (at low loads), sleeveless laminated chains.

The supporting devices and cars should ensure convenient, reliable and easy loading and unloading and steady position of load units along the whole length of a conveyer, i.e. at all working and assembling places. The supporting devices should be provided with fasteners, locks, clamps, etc., when needed, and also with mechanisms for tilting, turning or lifting of load units as required by the assembling process. The centre of gravity of a load unit should be always inside the supporting contour of a car in all assembling positions. If, by the conditions of assembling, load units on cars should have a cantilever position (for instance, aircraft wings and automobile chassis on assembling conveyers), the conveyer is provided with additional counter-rails or stops to ensure steady position of the units. Large, heavy and long load units are carried on double-chain conveyers with separate small cars fastened individually to each of the chains (Fig. 9.5). Wheels of conveyer cars are usually mounted on rolling bearings.

The drive is of a conventional type and connected to the drive sprocket through a reducer gear with a single additional chain or gear transmission (see Fig. 9.1). A chain transmission is employed when the drive is arranged in a pit under the floor so as not to block the production area and ensure easy removal of loads at the conveyer end. A gear transmission is used in the drive of conveyers with non-tipping cars and cantilever arrangement of drive sprockets.

Conveyers for continuous transportation of loads usually have a variable-speed drive which allows smooth variation of the speed of cars. In order to prevent conveyer damage on an occasional overload or jamming of the chain, the drive sprocket or transmission has a shear pin which is cut off as soon as the tractive force exceeds the allowable level (see Ch. 3) and the conveyer is stopped. Besides, tipping-car conveyers have an additional safeguard in the form of a limit switch Sw (Fig. 9.1*a*) interlocked with the drive motor; it stops the conveyer when a loaded car approaches a turning sprocket.

The take-up is of the conventional screw or spring-screw type; the latter is preferable in conveyers of a large length. In the take-ups for double-chain conveyers, one of the sprockets is keyed on the shaft and the other rotates freely to ensure self-adjustment of sprocket teeth, if the two chains have different length of chain pitch.

The travel of a take-up is usually 400-500 mm. With strong wear, both chains should be replaced at once so as to ensure a constant pitch of cars.

9.2.3. Conveyer Calculation

The main characteristics of a conveyer are chosen as recommended in Sec. 9.1 and Ch. 2. The load-carrying capacity of cars is calculated for the largest mass of load units on the conveyer, including the mass of various attachments, and the pitch of cars is determined by their length which depends on the longitudinal size of loads to be transported.

The rate of a process, i.e. the time (minutes) during which a single final unit of product passes from the conveyer, can be found by the formula

$$t_p = 60 T_{sh} m K_t K_l / Z_d \qquad (9.1)$$

where T_{sh} is the number of hours in a shift; m is the number of shifts per day; K_t is the coefficient of time utilization of the conveyer which considers probable standstills (usually $K_t = 0.9$-

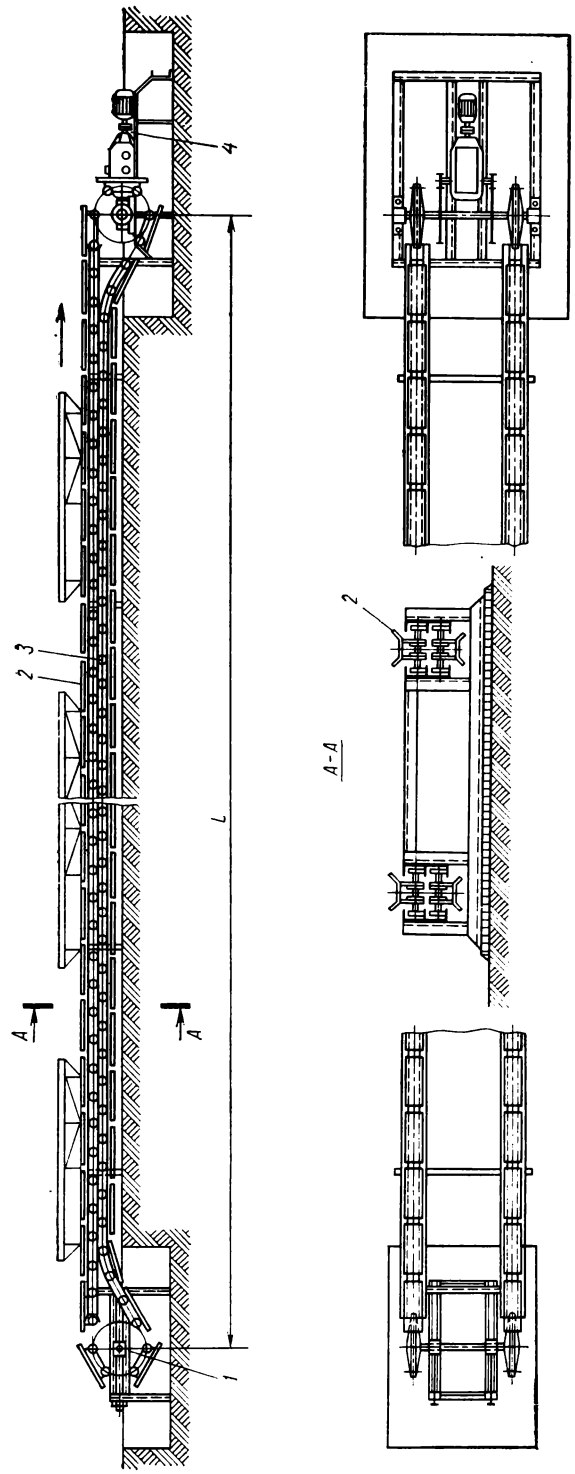


Fig. 9.5. Double-chain conveyer with separate cars for long load units (GDR)
1—take-up; *2*—car; *3*—hauling chain; *4*—drive

0.95); K_l is the coefficient of car loading on the conveyer (usually $K_l = 0.85\text{-}0.9$); and Z_d is the number of articles manufactured per day (daily productivity) on the conveyer line.

The time for performing an operation on a working place with conveyer continuous motion

$$t_w = t_p \tag{9.2}$$

With intermittent motion of the conveyer, when an operation is performed during conveyer standstill

$$t_w = t_p - t_m \tag{9.3}$$

where t_m is the time to move the conveyer to a next position.

The speed of a continuously moving conveyer

$$v = na_c/t_w \tag{9.4}$$

where n is the number of articles on a car, and a_c is the car pitch.

Assembly line conveyers usually have a speed of 0.6-6 m/min in continuous motion or 6-8 m/min in intermittent motion.

If the total labour consumption for assembly, T_Σ, and the operations are distributed uniformly in time, the number of working places at a conveyer (allowing for a certain reserve) will be

$$i = T_\Sigma/t_p \tag{9.5}$$

and the total length of a process line conveyer

$$L_p = a_c i \tag{9.6}$$

If a certain process (say, cooling, drying or testing) to be performed on the conveyer requires a time t (min), the length of this section with the adopted conveyer speed v (m/min), will be

$$l = vt \tag{9.7}$$

The pull calculation for vertically closed car conveyers is carried out by the conventional method, i.e. by passing over the conveyer path and determining the tensions in the characteristic points (see Ch. 2).

Linear gravity forces are determined from a curve of car loading in the production process, since the mass m_l of load units in the process may vary periodically (usually it increases, see Fig. 9.1*b*).

Preliminary calculations and the choice of a particular size of the tractive element can be checked by calculating the highest tension of conveyer chain $S_{\max}$ by the generalized formula

$$S_{\max} \approx \xi (S_0 + w \Sigma q_i l_i) \tag{9.8}$$

where ξ is the coefficient of local resistances on turning sprockets (usually $\xi = 1.08\text{-}1.12$ depending on the operating conditions); S_0 is the initial chain tension (usually $S_0 = 1\text{-}2$ kN); w is the coefficient of resistance of the moving part of conveyer (see Ch. 2); q_i is the linear load in various sections of conveyer path, N/m (the mass of cars is taken approximately by analogy with existing conveyers); and l_i is the length of individual sections of the conveyer with different linear loads, m.

9.3. Carrier-chain Conveyers

A carrier-chain conveyer has two vertically closed hauling chains (usually of laminated wheel-type), with stops (seats) fastened on chain plates with a certain pitch. The loads to be transported are laid on the seats (or on the top sides of widened chain plates). The chains move on wheels along guide tracks.

Carrier-chain conveyers may have a horizontal straight or combined path (Fig. 9.6). With a combined path, one or both end portions of a conveyer are made inclined to arrange the take-up and/or drive in pits under the floor, so as to ensure convenient loading and unloading of the conveyer and avoid blocking of the floor of production premises.

Carrier-chain conveyers are simpler in design than common car conveyers, but can only be used with load units of

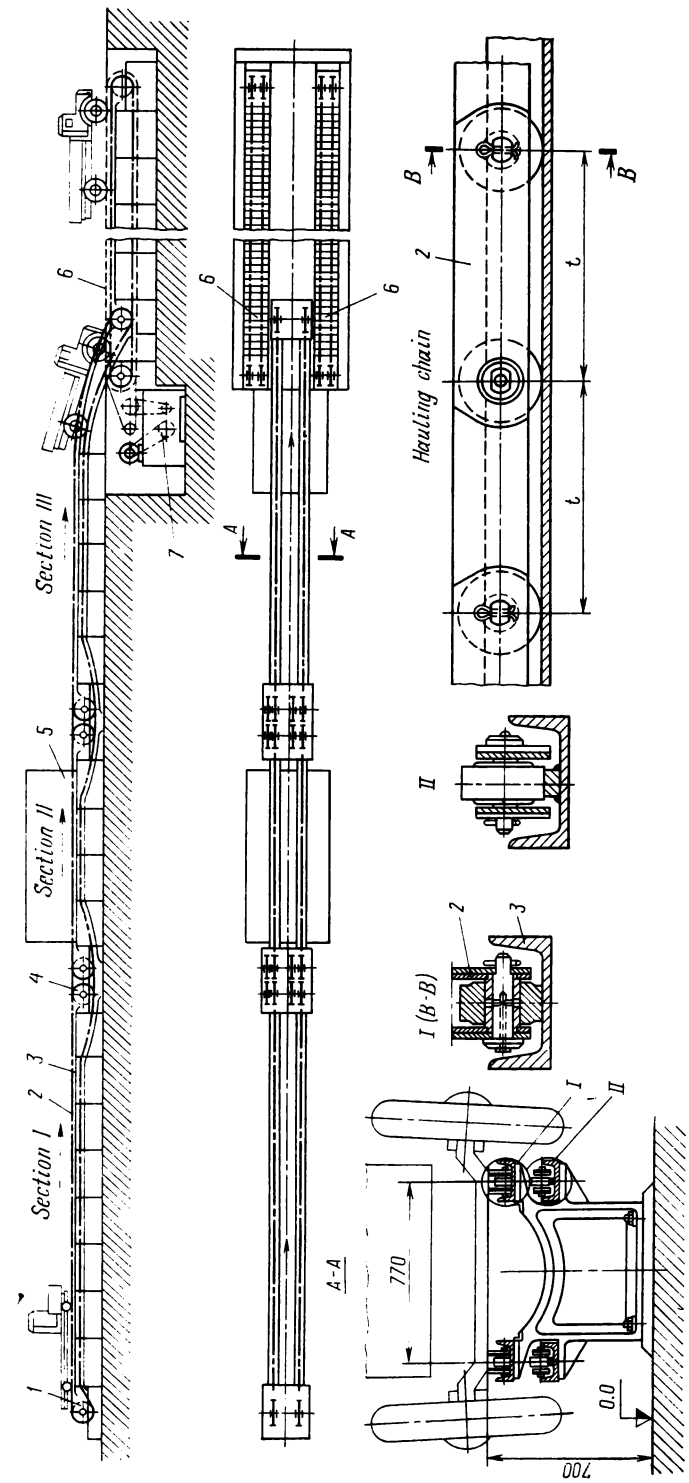


Fig. 9.6. Carrier-chain conveyer for assembly of lorries
1—take-up; *2*—hauling chains; *3*—guide tracks; *4*—transfer device; *5*—painting and drying boxes; *6*—apron conveyers; *7*—drive

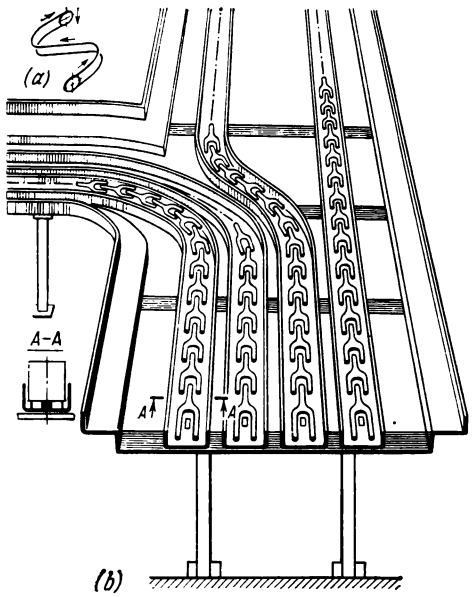


Fig. 9.7. Scheme of a multi-strand carrier-chain conveyer with clevis chains
(a) path of a single chain strand; (b) mid portion of the conveyer with sideways

a suitable (mostly flat) shape on the bottom.

Carrier-chain conveyers are employed in main assembly lines of automobiles, tractors and like machines, for conveying large machine parts between process operations, for conveying hot goods, and as the travelling bottom of some types of heating furnaces. In an automobile assembly line (see Fig. 9.6), the articles being assembled pass through a painting and a drying chamber, which are supplied by a separate conveyer section *II*. The conveyer is divided into three sections to avoid contamination of the whole conveyer by paint. The entire conveyer line is moved from a common drive. Individual sections are connected by chain transmissions which in sections *I-III* serve to transfer articles from one section to another.

Small uniform articles (such as bottles, cans and the like) are transported on single- or multi-strand carrier-chain conveyers with clevis chains (Fig. 9.7). Clevis chains (see Fig. 3.16*c*), either cast or stamped steel chains or plastic chains, have enlarged gaps in joints, i.e. an elevated flexibility, and can move in a conveyer along a spatial path. The chain is mounted in a narrow trough and slides on its bottom; the side walls of the trough serve as guides in horizontal turns of the conveyer path. Articles are placed directly on chain links and carried inside the trough.

Carrier-chain conveyers with flexible clevis chains have found wide application in lines for filling, washing, capping and sorting of bottles and cans and in other process lines.

The main characteristics of carrier-chain conveyers are as follows: chain speed 0.3-0.5 m/s; linear load 50-1000 N/m; coefficients of sliding friction of steel chain without lubrication $f = 0.40$-0.45 or, when lubricated with oil or water-soap emulsion, $f = 0.15$-0.2; the coefficient of resistance in horizontal turns $\xi = 1.1$ at a turning angle of 15° and $\xi = 1.5$ at 90°.

At high loads, the chains may be provided with supporting wheels.

The calculation of carrier-chain conveyers is done as for vertically closed car conveyers (see Sec. 9.2) or spatial-path overhead trolley conveyer (see Ch. 8).

9.4. Horizontally Closed On-floor Car Conveyers for Casting Moulds

9.4.1. Design

Foundry shops mainly employ the following three types of on-floor car conveyers: conveyers with separate car platforms for conveying casting moulds on a horizontally closed plane path (type Г, Fig. 9.8) or a spatial path (type П, Fig. 9.9) and conveyers with overlapping platforms (continuous apron) and plane path (type ГС).

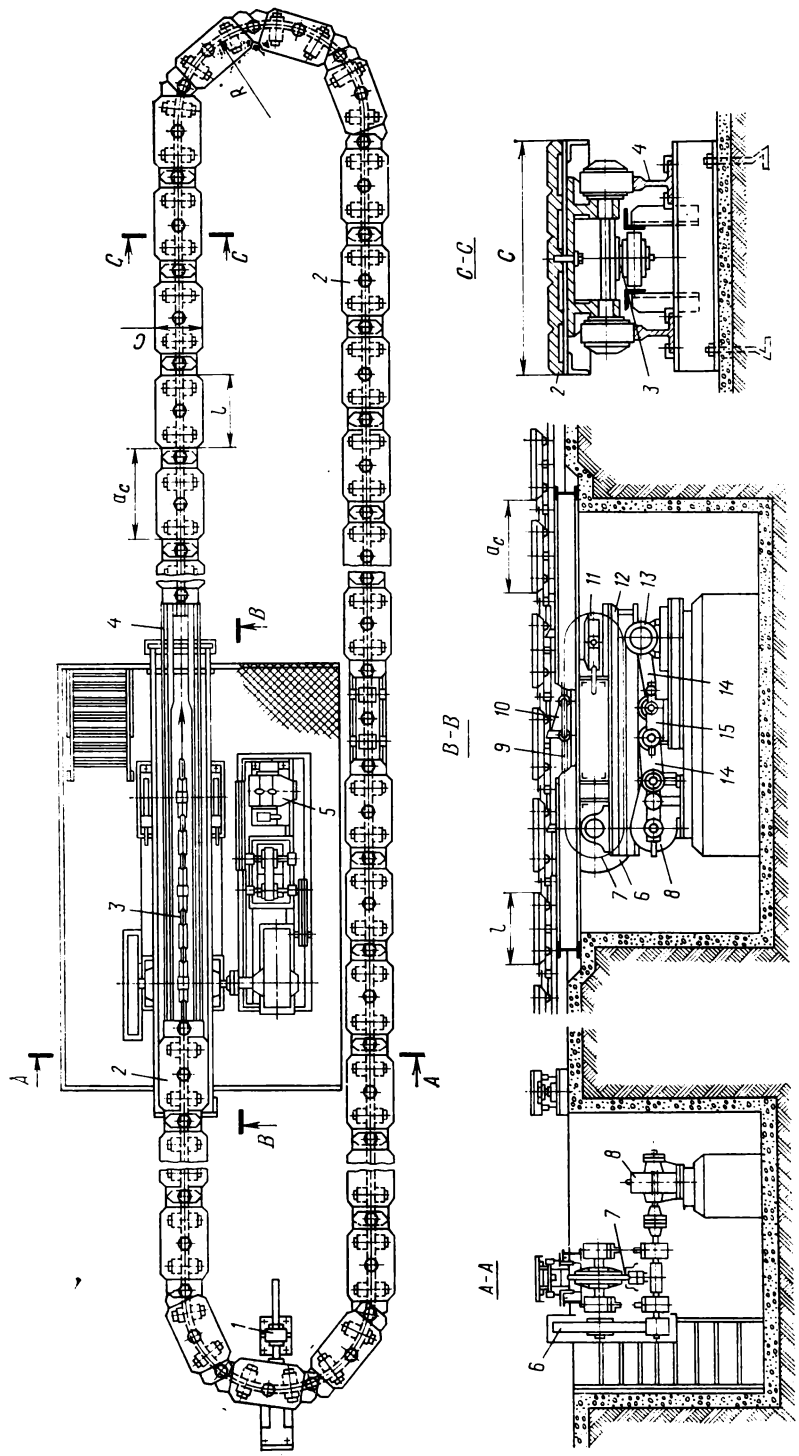


Fig. 9.8. Horizontally closed plane-path car conveyer type Г for casting moulds (USSR)

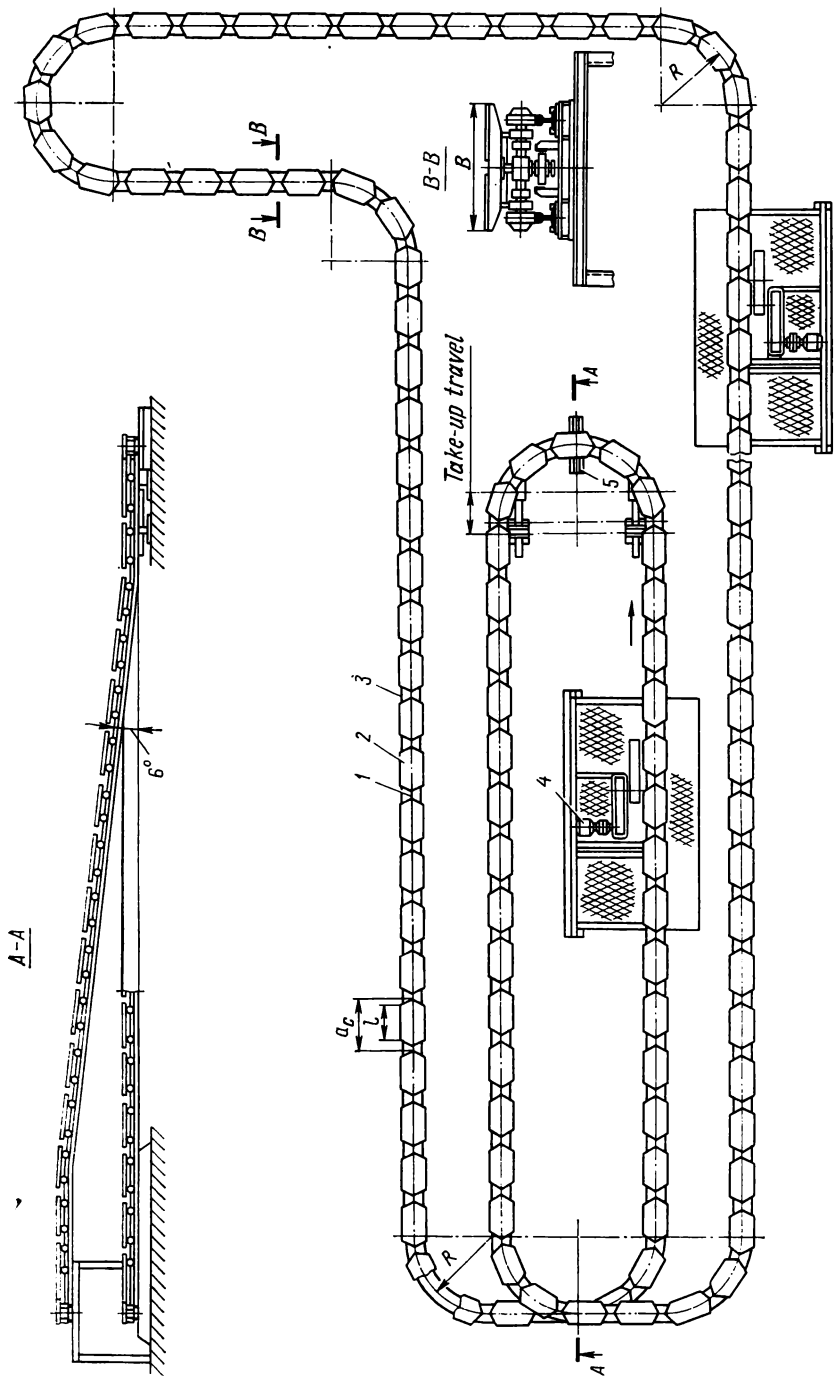


Fig. 9.9. Spatial-path car conveyer type II for casting moulds (USSR)

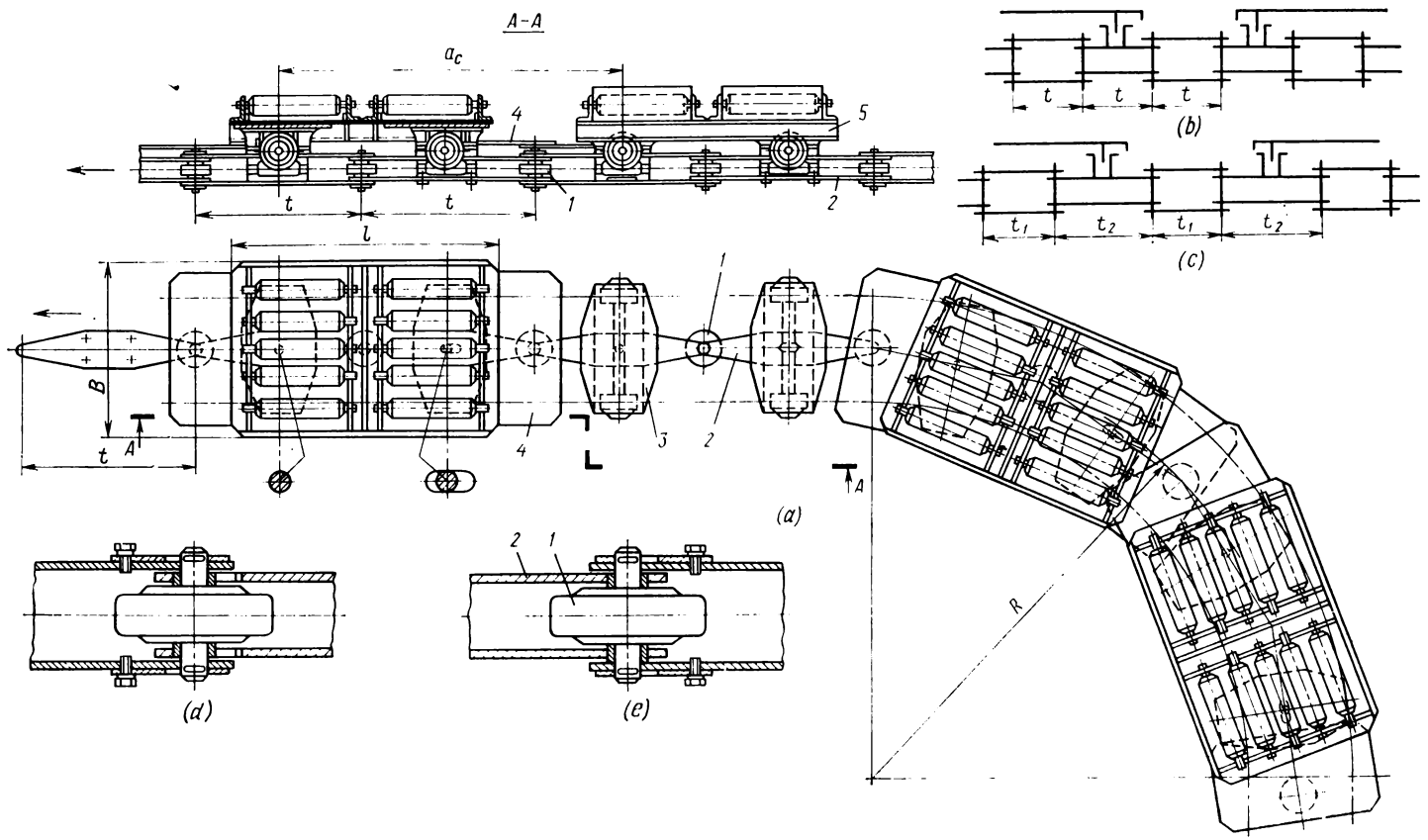


Fig. 9.10. The moving part of car conveyer type Г for casting moulds
(*a*) general view; (*b*) scheme of joining of chain links with even pitch; (c) scheme of joining of chain links with different pitch; (*d*) hauling chain joint between platforms; (*e*) hauling chain joint under the platform

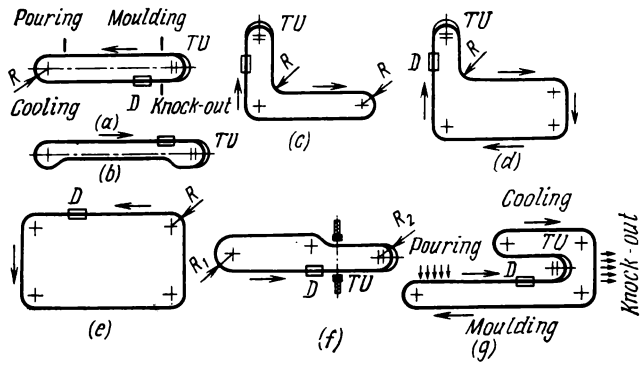


Fig. 9.11. Various path configurations of horizontally closed car conveyers, types Г and ГС
D—drive; *TU*—take-up

A horizontally closed on-floor car conveyer, type Г (Fig. 9.8), has a centrally arranged hauling chain *3* (with vertical axes of joints), car platforms *2* which are fastened to the chain and move on two guide tracks *4*, a drive *5*, and take-up *1*. The moving part of the conveyer (Fig. 9.10) consists of a laminated wheel chain *2*, single-axle cars *3* rigidly fastened to the chain plates, platforms *5* to carry casting moulds, and shields *4* to protect the chain against fouling with foundry sand and metal splashings. Each platform is supported by two cars and fastened to them by means of pins which enter holes in the cars. The hole in one (the leading) car is round and that in the other is elliptical to compensate for variations in the spacing between the cars in turned portions of the path.

Wheels with ball or tapered roller bearings are mounted on pins of the hauling chain (see Fig. 3.7); they move between guide bars which prevent the cars from moving aside from the track.

A section of the hauling chain for a platform is composed of two or four chain links; in four-link sections the pitch of links may be different (for instance, 250 mm and 400 mm), depending on the pitch of platforms (Fig. 9.10*b* and *c*). In chain sections between the platforms, one of the joints is made movable by providing elongated holes at one side of internal chain plates (Fig. 9.10*d*). A chain with movable joints can contract under the action of compressive forces and vary its length in horizontal turns of the conveyer track.

Since conveyers of this type have single-axle cars with flangeless wheels and hinged joints between the cars and platforms, the moving part of the conveyer can pass through horizontal turns of relatively small radii, and therefore, the paths of conveyers may be of diverse configuration in plan (Fig. 9.11).

The total length of the conveyer path is determined as the sum of the lengths of its individual sections in various zones of the foundry shop (Fig. 9.11). In particular, the moulding and assembling sections have usually a length around 30-50 m, casting section 10-15 m, cooling section 40-60 m, and knock-out section 3-10 m.

Conveyers of this type are easily adaptable to various path configurations and various production conditions, have a high reliability, allow mechanized loading and unloading, and make it possible to utilize the entire length of the path.

Conveyers with separate cars and spatial path, type П (see Fig. 9.9), have essentially the same design as type Г, the only difference being that the hauling chain has an additional joint with horizontal pin *1* (Fig. 9.12) to make the moving part flexible in the vertical plane. Spatial-path conveyers make it possible to utilize more rationally the production area and improve sanitary conditions in foundry

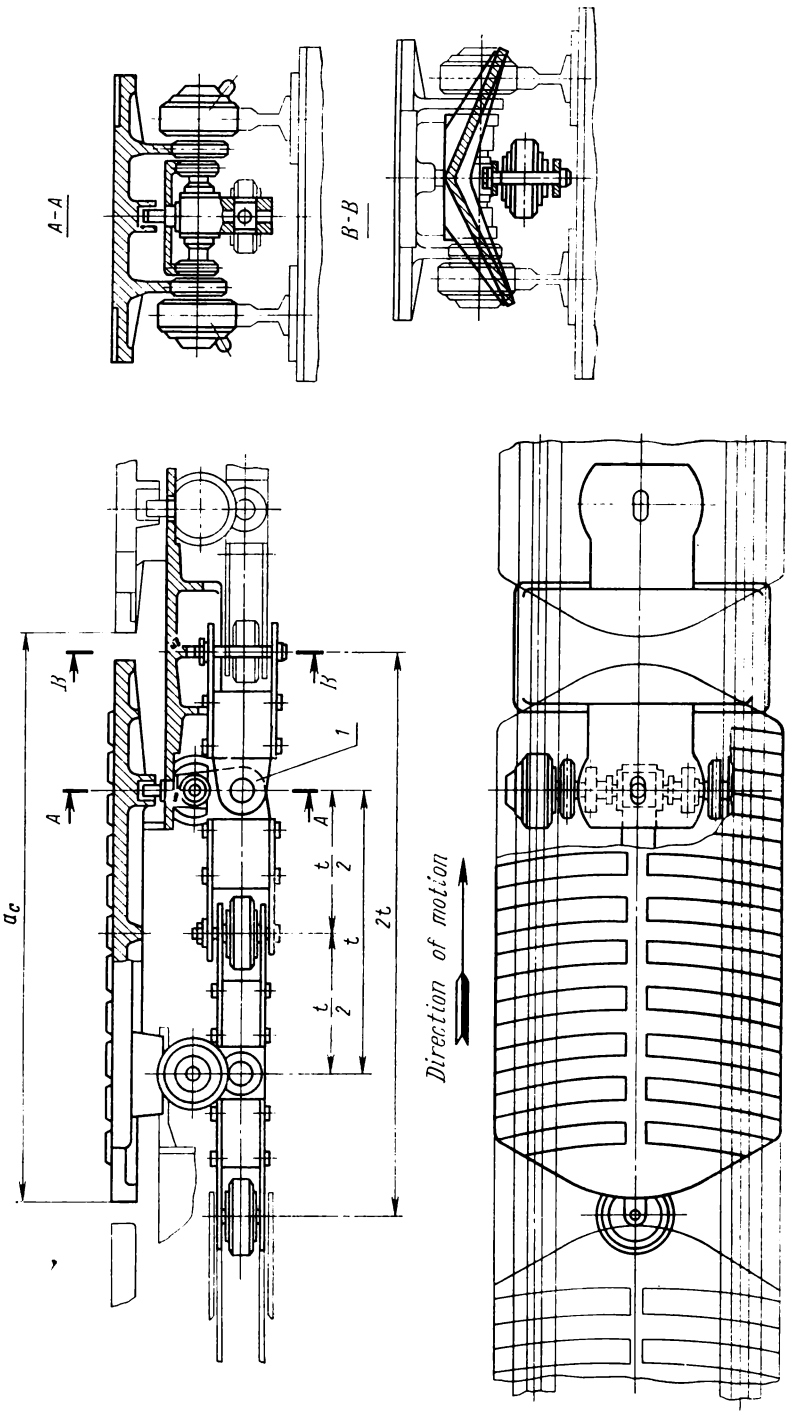


Fig. 9.12. The moving part of car conveyer, type II (USSR)

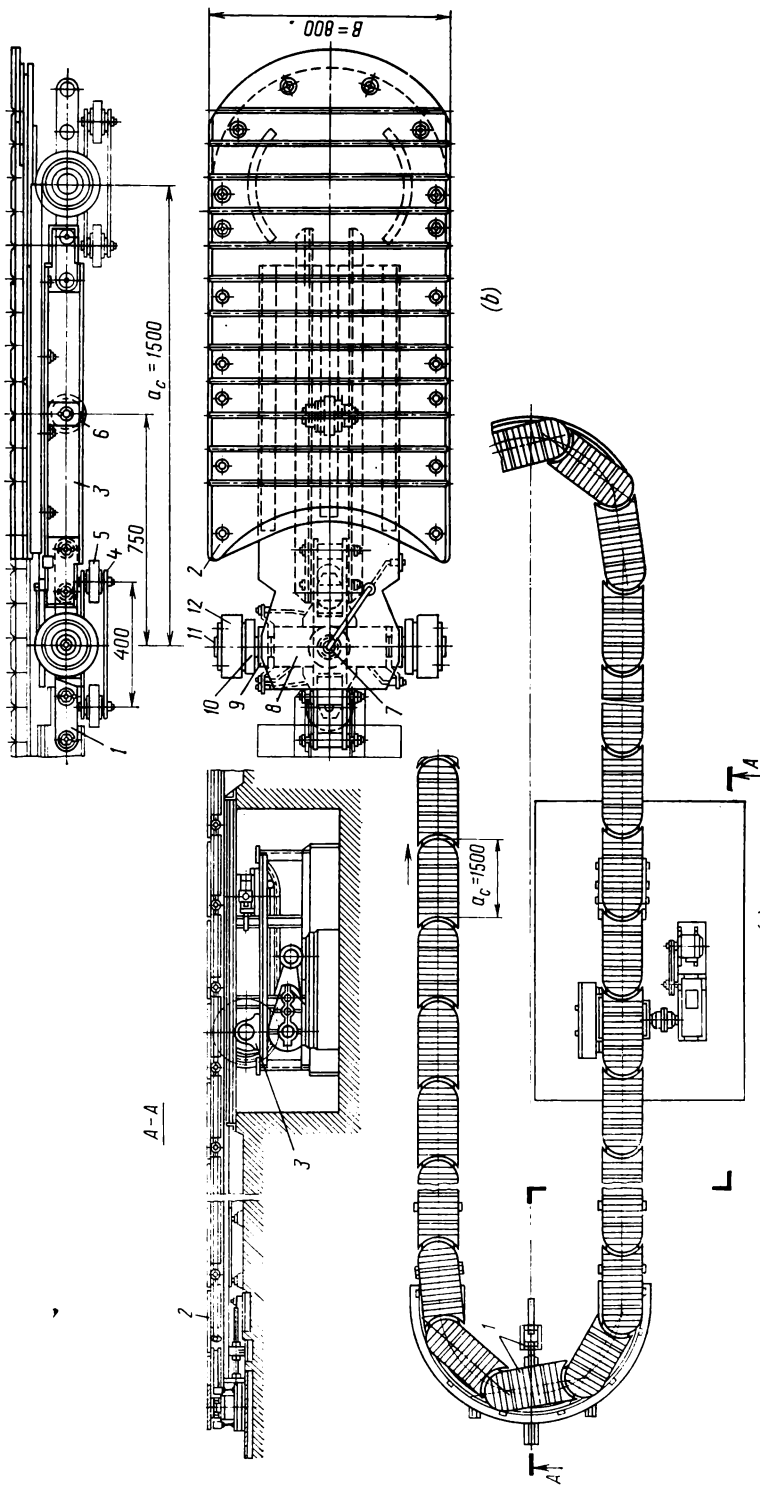


Fig. 9.13. Horizontally closed car conveyer, type TC (USSR)
(a) general view; (b) moving part

shops, when the moulding and casting sections are arranged on the first floor and the cooling and knock-out sections beneath. They may have vertical bends of a radius of 5-8 m and the highest angle of incline, 6°.

In car conveyers, type ГС, for the transportation of casting moulds (Fig. 9.13), car platforms have semicircular ends (convex at one end and concave at the other) and form a continuous apron both in straight and curved portions of the path, thus protecting the cars and tracks from fouling by sand and metal splashings. The conveyer has no hauling chain and the functions of the load-carrying and tractive element are performed by car platforms.

A section of the moving part of conveyer (Fig. 9.13*b*) comprises a car *8*, platform *2*, longitudinal beam *3*, and connecting links *1*. The car has a horizontal axle *11* with wheels *12* and platform-supporting rollers *10* and *9*, a vertical axle *7* for hinged connection of the longitudinal beams *3* of adjacent sections, and a rocker *4* with horizontal wheels *5*. Rollers *10*, which are mounted closer to the longitudinal axis of the platform, support the platform of their own section, whereas rollers *9* support an adjacent platform. Support rollers diminish friction losses during turning of platforms. The rocker with wheels *5* guides the cars in conveyer turns. The conveyer has a caterpillar drive and a spring-screw take-up. The beam *3*

Table 9.1. Main Characteristics of Car Conveyers for Casting Moulds (see Figs. 9.8 and 9.9)

Platform dimensions, mm		Car pitch a_c, mm	Mass of 1 m of moving part with flat-floor platforms, kg/m	Load-carrying capacity of platform, t	Maximal turning radius R_h in horizontal plane, m	Coefficient of resistance of cars in straight sections w	Recommended car speeds, m/min
width B	length l						
500	650	800	220	0.5	1.25	0.025	3; 4.8; 7.5
	800	1000	200				
	1000	1250	180				
650	650	800	260	0.5	1.6	0.020	1.9; 3; 4.8; 7.5
	800	1000	225-320	0.5-1			
	1000	1250	210-290	0.5-1			
	1250	1600	250	1			
800	800	1000	340	1	1	0.020	1.9; 3; 4.8; 7.5
	1000	1250	320-380	1-2	1-2		
	1250	1600	280-340	1-2	1-2		
	1600	2000	320	2	2		
1000	1000	1250	430	2	2.5	0.015	1.2; 1.9; 3
	1250	1600	370-780	2-4			
	1600	2000	350-690	2-4			
	>2000	2500	590	4			
1200	1250	1600	860	4	3.15	0.015	1.2; 1.9
	1600	2000	780-980	4-8	3.15		
	2000	2500	680-880	4-8	3.15		
	2500	3150	790	8	2.5		

has a roller *6* in the middle, which engages the pusher of the caterpillar drive.

The main characteristics of car conveyers for casting moulds are given in Table 9.1; the highest tension of the hauling chain is 30-50 kN and the highest speed, 12 m/min.

9.4.2. Conveyer Elements

The tractive element in horizontally closed car conveyers for casting moulds is a single laminated wheel chain with an elevated mobility of some joints (see Fig. 9.10*d*).

In spatial-path conveyers (see Fig. 9.9), the tractive element is made in the form of a double-hinged chain with vertical and horizontal joints which make it flexible vertically and horizontally, or in the form of sections of a common laminated chain which are attached to cars by joints whose axes are perpendicular to the axes of chain joints. Car platforms of spatial-path conveyers are provided with stops to prevent sliding of loads on inclined sections.

Car platforms for conveying casting moulds are known in three versions (Fig. 9.14): with a flat floor, with a roller-table floor, and a tilting type. Flat floors of platforms are made of cast iron; they have ribs on the underside to prevent buckling, and grooves on the top to allow free exit of gases from filled moulds. Flat-floor platforms are mainly employed for heavy moulds, if these can be loaded and unloaded by an overhead crane or electric hoist, and for light moulds to be loaded and unloaded manually. Roller-table platforms are used for moulds of a mass up to 2000 kg which are loaded from and unloaded onto the roller table by means of a pusher mechanism. A roller-table floor of platforms is made in the form of two sections which are pivotedly fastened to the platform frame and can be turned aside to remove fallen-through sand from beneath the rollers.

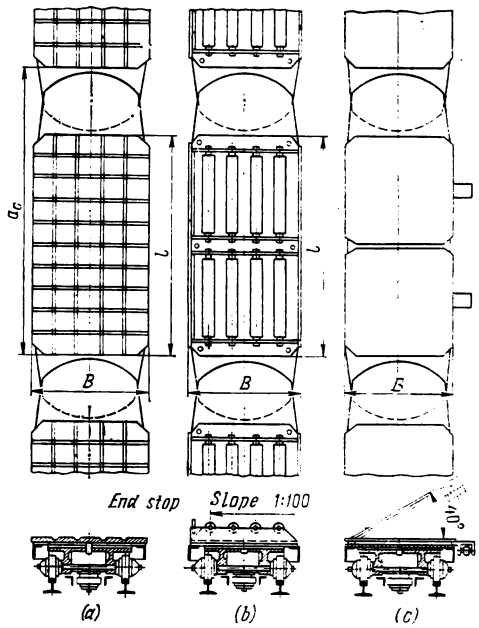


Fig. 9.14. Platforms of car conveyers for casting moulds
(*a*) with flat floor (for conveyer types Г, ГС, and П); (*b*) with roller-table floor (for type Г conveyers); (*c*) with tilting platform (for type Г)

Tilting-type platforms, of a width up to 650 mm, are employed in permanent-mould foundry shops. They are unloaded automatically by tilting the platform sideways. In a moving conveyer, the roller at the side of a platform runs onto a curved bar, so that the platform is tilted aside (see Fig. 9.4) and a mould slides from it onto a knock-out grid. Car wheels of the conveyer are made of cast iron СЧ15 by chill casting or, more often, of steel 45 with induction hardening of the rim, and mounted on rolling (ball or tapered roller) bearings to diminish the required tractive force. In view of heavy operating conditions in foundry shops, wheel bearings should be reliably protected by labyrinth seals.

The drive is of the caterpillar type (see Figs. 9.8 and 9.15) and consists of a vertically closed endless driving chain *9* (see Fig. 9.8) with knuckles *10*, drive sprockets *7*, take-up sprockets *11*, gear transmission *6*, reducer gear *8*, variator *15*, V-belt transmissions *14*,

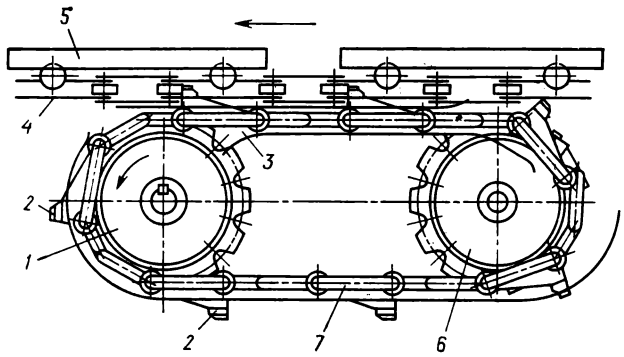


Fig. 9.15. Caterpillar drive for car conveyers
1—drive sprocket; *2*—knuckle; *3*—guide bars; *4*—hauling chain; *5*—platform; *6*— take-up sprocket; *7*—driving (caterpillar) chain

electric motor *13*, and metal structure *12*. The pushers (knuckles) of the driving chain (Fig. 9.16) engage the wheels of the hauling chain of the conveyer. For easy disengagement of knuckles from wheels, the pitch of knuckles should be smaller by 3-5 mm than the pitch of wheels of the conveyer chain, so that a next knuckle comes into engagement before the previous one is disengaged. This ensures smooth transmission of the tractive force from the driving chain to the hauling chain of the conveyer.

To prevent the drive mechanisms from breakage on occasional overload, the gear wheel on the drive shaft has a shear pin (see Ch. 3). The drive is usually located in a conveyer after the most heavily loaded section, for instance, after the cooling section before the knock-out section.

Standard drives are equipped with electric motors for 1.5-15 kW and have a spacing of 2010-2675 mm between the sprockets.

The take-up (Fig. 9.17) is arranged in 180°-turn of the conveyer. It has a strong frame *4* with curved rail sections *2* and guide tracks *1* fastened to it. The frame is placed on supports *3* and can move on them by means of a tensioning screw *5*. The take-up travel is usually 500-600 mm. The ends of the sections of rails and tracks are connected with the respective stationary elements of the conveyer by means of extensible joints.

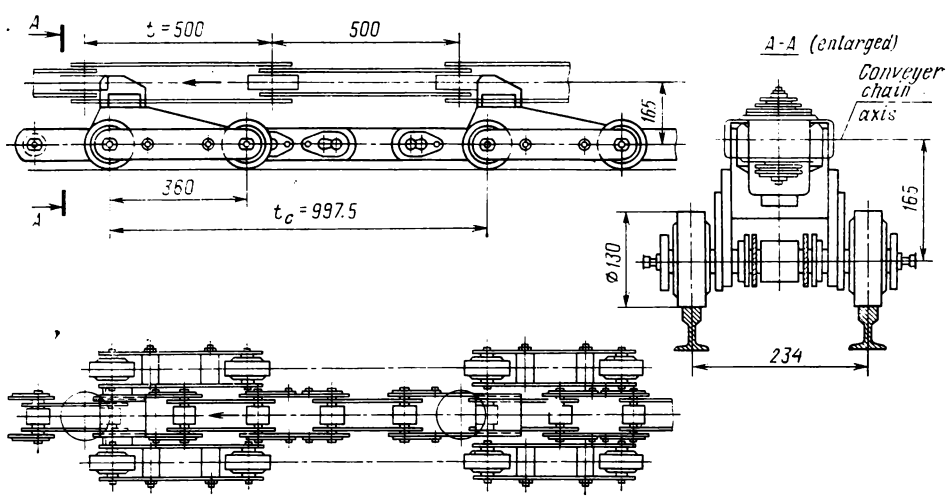


Fig. 9.16. Driving chain with knuckles

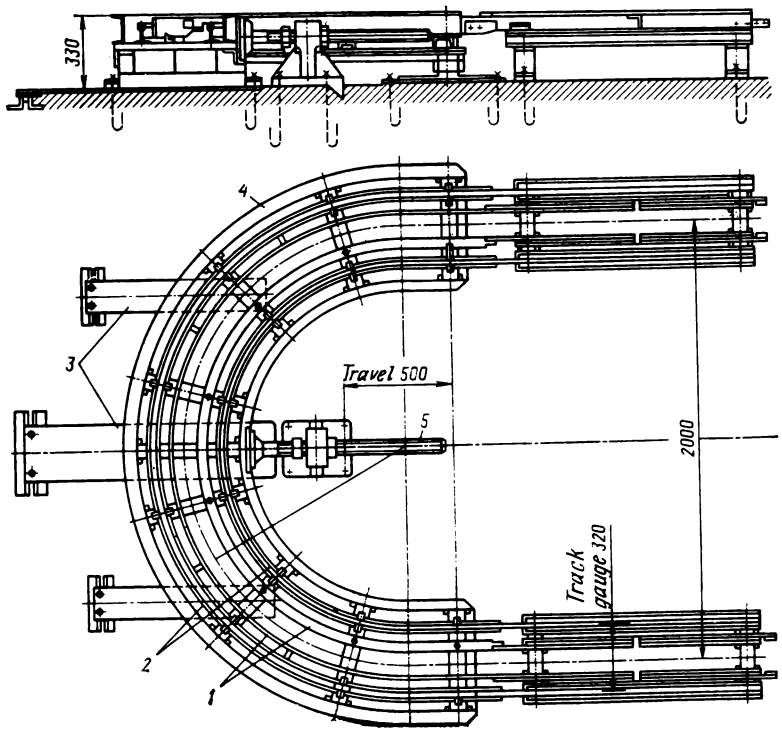


Fig. 9.17. Take-up of a horizontally closed car conveyer

Take-up devices are indispensable in conveyers with an intricate horizontal or spatial path (see Fig. 9.11*b-d*, *f*, and *g*) and in those having an appreciable length (more than 50 m). Low-speed (v less than 4 m/min) conveyers with a simple path (see Fig. 9.11*a*) and small length (up to 50 m) may have no take-ups, provided that their elements are manufactured and assembled with a higher accuracy. Take-ups are not employed in conveyers with a rectangular path configuration (see Fig. 9.11*e*), since they would complicate largely the conveyer design.

The tracks of conveyers comprise supporting and guiding sections. The former are made from railway rails types P11, P24 or (for heavy-duty conveyers) P33. The guide tracks, between which the hauling chain wheels move, are made from steel angles (usually 60 × 60 × × 6 mm or 75 × 60 × 6 mm). The tracks are fastened by bolts to welded or cast support shoes.

9.4.3. Conveyer Calculation

The dimensions of car platforms are chosen according to the size and mass of loads and usually are by 100-150 mm greater than the overall dimensions of the moulds. The dimensions of the car platforms determine their pitch which must be a multiple of the length of chain sections. The average rated speed of a conveyer for assembling operations is determined by the rate of these operations and the required throughput capacity, considering the coefficients of loading and time utilization of the conveyer. For an adopted speed (usually within 1.2-7.5 m/min, see Table 9.1), the lengths of various processing sections and the time of presence of loads

on them are checked and corrected by using formula (9.7).

The pull is calculated in the conventional order by summing up the resistances in various sections of the conveyer path. The mass of the moving part is taken according to Table 9.1. The initial tension is taken equal to 1-1.5 kN. Resistances in various sections of the conveyer path are determined as recommended in Ch. 2.

The theoretical coefficients of resistance of cars in straight sections of conveyers for casting moulds (with car wheels mounted on tapered roller bearings) are given in Table 9.1.

The resistances in curved conveyer sections are found depending on the type of cars and turning device employed, separately for the cars and chain. The resistances in vertical bends of spatial-path conveyers are calculated in the same way as for overhead conveyers (see Ch. 8).

Horizontally closed on-floor car conveyers are also employed for transportation of other diverse loads and articles (for instance, gas ovens, refrigerators, etc.) and in various manufacturing processes (assembling, washing, painting, etc.). Their designs are quite diverse, but they are substantially simpler and lighter than foundry conveyers. These conveyers may have a central or side arrangement of the hauling chain, with single- or double-rail tracks, and with a chain-and-sprocket or caterpillar drive. Horizontally closed car conveyers are known which have no hauling chain and are driven from a rotating tube arranged stationarily along the path of car motion. Each car has a turning guide disc which contacts the drive tube. The car is moved due to friction forces appearing between the disc and tube. The speed of cars is controlled by turning the disc relative to the tube axis. When the disc is perpendicular to the tube axis, the car remains in place. The motion of cars is controlled by guide bars which vary the positions of their guide discs.

9.5. Distributing Table-type Car Conveyers with Automatic Addressing

9.5.1. Design

In a distributing table-type car conveyer (Fig. 9.18*a* and *b*), the cars are fastened to the hauling chain and move on guide tracks of the supporting metal structure at the level of working places (usually 700-800 mm above the floor). The goods to be conveyed (clothes, footwear, parts of instruments, radio and television sets, small machine parts, etc.) are placed on the cars directly or in open box containers. Loading may be performed manually or automatically. With automatic loading, filled containers are moved from a working-place table onto cars automatically by a pneumatic or mechanical pusher. The mass of load units (or filled containers) is usually 0.1-25 kg.

The cars of a conveyer (or sometimes the containers) are provided with address carriers for automatic addressing of loads and their unloading at particular working places which have address readers and receiving tables for the loads. The supporting plates of cars are made turning or tilting and have controlling stops; these are needed only for loads conveyed in containers. When the address set on the address carrier coincides with that set in the reader at a working place, the car plate is turned or tilted by means of a controlled guide bar and the load slides onto the working place.

A conveyer may have a centralized control of load addressing (from a central control panel) by a programme or local control from working places. In the latter case, as a worker has finished his operation, he places the article on a free car and selects the address for further motion of the article. Conveyer cars may move continuously (usually at a speed of 0.5-6 m/min) or intermittently (8-10 m/min)

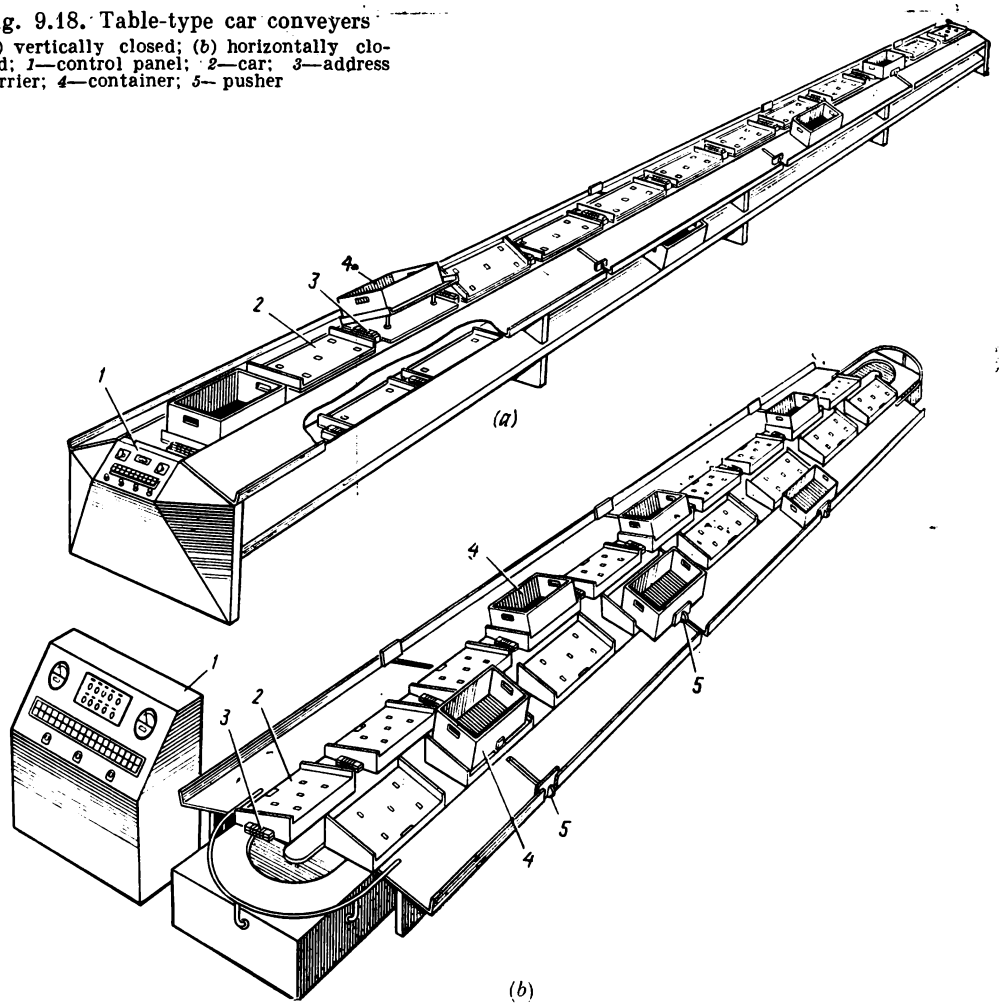


Fig. 9.18. Table-type car conveyers
(*a*) vertically closed; (*b*) horizontally closed; *1*—control panel; *2*—car; *3*—address carrier; *4*—container; *5*— pusher

and their speed is controlled from the central panel.

As regards the path configuration and location of the hauling chain, distributing conveyers may be vertically closed with non-tipping or tipping cars or horizontally closed (Fig. 9.18*b*) or may have a spatial path.

Distributing car conveyers are known in various designs, depending on the processes in which they are employed (Fig. 9.19).

Distributing car conveyers are intended for the transportation of loads and articles between working places in flow, mass and series production in the light (sewing, shoe-making, etc.), radio-electronic, instrument-making and machine-building industries. Distributing conveyers possess certain favourable features, in particular, offer wide possibilities for automatic addressing, which makes them suitable also in applications when there is no strict sequence in process operations or for delivery of materials and articles to a number of working teams which do their work at different rate. In the last case, working operations are performed not directly on the conveyer (as is the case in assembling-line conveyers), but on the working places near the con-

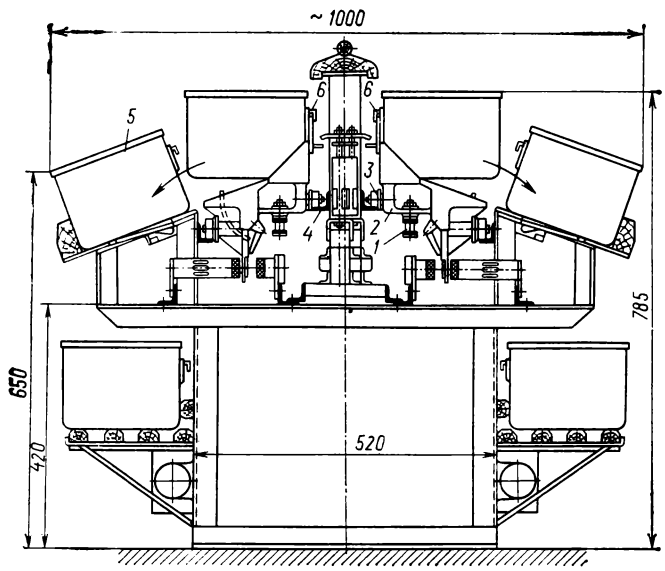


Fig. 9.19. Cross-sectional view of a semi-automatic horizontally closed table type car conveyer for footwear
1—hauling chain; *2*—car; *3*—car wheels; *4*—tracks; *5*—container; *6*—container-holding lock

veyer and the distribution of articles is controlled by the worker or dispatcher to a predetermined programme. For that reason, conveyer cars with loads can circulate over the conveyer path and supply (unload) them to any desired working place.

The main characteristics of distributing car conveyers are as follows: load-carrying capacity of cars 10, 25 or 50 kg; width of platforms 250-300 mm; and length 400-800 mm. Loads are usually carried in standard containers.

A distributing and sorting spatial-path car conveyer (Teleflex Co., France, Fig. 9.20) has cars *3* attached to plates of a double-hinged chain *5*. The load-carrying portion of a car has two shelves *1* which can tilt on shaft *2* upon a control action on guide wheel *4*. The cars move on tracks *6* and the chain, on tracks *7* made from thin bent profiles. At loading and unloading stations, a wheel *4* contacts a guide bar and is deflected, so that the shelf *1* connected with it is tilted to receive a load or slide off the load into a receiver. Loading and unloading are automatically controlled. The conveyer is loaded by means of automatic rotor-type loaders.

The conveyer has been developed for conveying and sorting of parcels, bales, bags, packages and other uniform unit loads.

The characteristics of these conveyers are as follows: load-carrying capacity

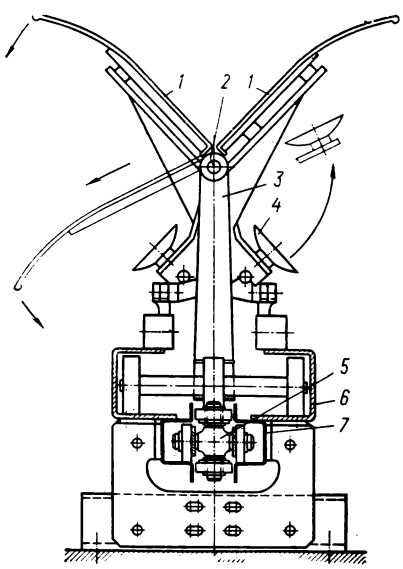


Fig. 9.20. Cross-sectional view of a car conveyer with automatic loading and unloading (Teleflex Co., France)

of cars 10, 30 or 70 kg, with the car length respectively 600, 900 or 1200 mm. Owing to wide application of plastics in the elements of chain and cars, the speed of cars attains 150 m/min. The chain has a pitch of 203 mm and can pass through turns of a radius of 0.6-1.8 m; the drive is of angular or caterpillar type, with the tractive force of 8-15 kN.

9.5.2. Conveyer Elements

In the main types of distributing car conveyers, the **tractive element** is made in the form of a single chain: laminated roller chain with a pitch of 80-100 mm; double-hinged chain with a pitch of 160-200 mm; or sometimes driving laminated chain with a pitch of 25.4-38.1 mm; the use of round-link chains is also known. Vertically closed conveyers with non-tipping cars have two standard hauling laminated chains. Double-hinged chains are employed in horizontally closed conveyers with an intricate path and especially in spatial-path conveyers.

The cars may have a smooth horizontal or tilting floor or a roller table. The frame of the car is rigidly attached to a chain link and is sometimes provided with an additional guide wheel (see Fig. 9.19). If wheeled double-hinged chains are used in a conveyer, the cars are replaced by wheelless platforms carried on the chains.

The take-up is usually of the screw or spring-screw type.

The drive is mostly of the reducer type, essentially of the same design as the angular drive of overhead conveyers, with the upper position of the drive shaft and the drive sprocket located above the reducer. In conveyers with intermittent motion of cars, the drive has a shoe brake for accurate stopping of cars at desired places.

The metal structure of the mid portion of a conveyer is made from rolled sections and tubes, and protected at sides by sheets to ensure the required safety. The structure is often equipped with height-adjustable seats for workers, with steps, and loading-unloading tables at the height of working places.

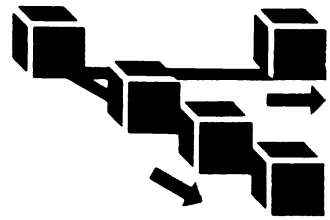
The automatic addressing system on a distributing conveyer sends loads to the desired unloading points and also performs counting and sorting of loads. Addressing systems exist in various designs, the most popular being the contactless system with local control.

Conveyer cars are unloaded by tilting their platforms or by sinking a pin which retains the load on an inclined platform.

Upon unloading, an empty car resets the actuating mechanism into the working position. The conveyer has an automatic locking system which prevents unloading of a car on a loaded place. If the given address is occupied, the car will not be unloaded, but propelled to a next working place with the same address or will perform a whole cycle on the conveyer path.

Distributing conveyers are calculated in the same way as other types of car conveyers (see Secs. 9.2, 9.4 and Ch. 2).

10 Load-towing and Walking-beam Conveyers



10.1. Principal Types

The load-towing (or simply, tow) conveyer is a conveyer type in which the tractive element moves unit loads on their own wheels (Fig. 10.1), on cars moving on the floor or guide tracks, by sliding the load units on a smooth floor, or by rolling on stationary unpowered rollers. In this type of conveyer, the car is not attached to the tractive element, but is moved by it with pushers or clamps. In this respect, tow conveyers differ principally from load-carrying car conveyers discussed in Ch. 9.

A tow conveyer has the tractive element in the form of single or, less frequently, two chains, steel cable, a combination of a chain with one or two rigid bars, or only one or two bars. Tow conveyers with the tractive element as a rigid bar or a combination of bar and chain or cable are called push-bar conveyers. A special group of push-bar conveyers includes flight-bar conveyers for bulk loads and special push-bar flight conveyers for curled metal chips (see Ch. 6). Strictly speaking, they do not relate to load-towing conveyers and for that reason are not discussed in this chapter.

As regards the position of the hauling chain, load-towing car conveyers can be divided into vertically and horizontally closed and those having a spatial path. The hauling chain may move on the floor of a building, under the floor, or on an overhead track. Accordingly, it is distinguished between tow conveyers with on-floor, under-floor and overhead position of the chain. The last group has been described in Sec. 8.5.

Tow conveyers are simple in design and inexpensive, have small dimensions, do not block production areas (except for conveyers with on-floor chain which, however, have only a limited application), and offer wide possibilities for automatic control. Tow conveyers with a spatial path and automatic addressing of cars can carry loads without overloadings on an intricate path to various floors of a building. Load-towing car conveyers are easily adaptable for interaction with on-floor periodic transport means: truck loaders, electric trucks, automobiles, piler cranes, etc. All loading and unloading operations can be performed on a conveyer in combination with these machines in a single production-and-transport cycle.

Among the drawbacks of tow conveyers are certain complications in maintenance if the chain, trucks and other conveyer equipment are arranged under the floor and the necessity to return empty cars in vertically closed conveyers.

Tow conveyers are employed for trans-

portation of unit loads in assembling, finishing, repairing, distributing and other transport-process flow lines in machine-assembling shops, repair depots, and mechanized stores. These applications are also typical of walking-beam conveyers which have no cars and move loads on a horizontal reciprocating frame.

Tow conveyers in assembling and other process operations may have continuous or intermittent motion and their speed is dictated by the rate of process operations and the spacings between working places (see Ch. 9).

10.2. Vertically Closed Tow Conveyers

10.2.1. Design

A vertically closed tow conveyer without cars (see Fig. 10.1) or with cars (Fig. 10.2) has a hauling chain (*3* in Fig. 10.1 or *4* in Fig. 10.2) which bends around a drive sprocket (*1* in Fig. 10.1 or *2* in Fig. 10.2) and a take-up sprocket (*8* in Fig. 10.1 or *5* in Fig. 10.2) and moves on a guide track *7* by sliding (see Figs. 10.1 and 10.2) or by rolling on wheels *2* (Fig. 10.3). The conveyer further has a support structure *6* (see Fig. 10.1), a drive (*2* in Fig. 10.1 or *1* in Fig. 10.2), take-up *9* (see Fig. 10.1), and turning devices *4*.

The load (tractor, railway car, automobile, aeroplane, etc.) is moved on its own wheels or on a car *3* (see Fig. 10.2) on tracks *10* (see Fig. 10.1) laid directly on the floor, i.e. beyond the conveyer structure, which simplifies largely the design of the conveyer proper. If moved on its own wheels, the load is attached to the hauling chain by hooks or ties *5* (see Fig. 10.1) which can be disengaged at the conveyer end by hand or automatically. If loads are conveyed on cars (or slide on the floor or roll on rollers), the hauling chain is provided with pushers (*6* in Fig. 10.2 or *1* in Fig. 10.3) which engage clamps (*2* in Fig. 10.2 or *3* in Fig. 10.3) of the cars or directly the stops provided on the loads. A clamp has a one-sided hinged fastening to the car frame and is tilted flush when the car is moved to a chain pusher.

The drive and take-up of a conveyer are arranged in a pit under the floor so as to allow free access for placing and unloading of goods from the conveyer and for removal of cars. The loads are loaded and unloaded by hoisting means. Self-moving loads (say, automobiles) can be removed from the conveyer on a slide on their wheels. A limit switch is usually provided after the last working place, which de-energizes the conveyer if the load has not been taken from it.

Vertically closed tow conveyers are employed widely in assembling and repairs of railway cars and flat-cars and their elements: undercarriages, frames, and bodies. A separate group is formed by vertically closed tow conveyers for transportation of loads on roller tables (Fig. 10.4) or a continuous floor; they are alternatively called pusher-chain conveyers and employed for conveying of casting moulds in foundry shops, pushing of ingots in heating furnaces, etc.

10.2.2. Conveyer Elements

The tractive element is a single chain of any type, most often laminated sleeve-type or sleeveless chain with a pitch of 80-200 mm or sometimes a steel cable 15-22 mm in diameter. The pushers are hingedly or rigidly fastened to the chain with a pitch according to the required spacing between loads. A tractive steel cable has hooks or clamps attached to it, which are connected to ties to tow self-moving (wheeled) loads.

Guide tracks are made from channels (see Fig. 10.1), angles or narrow-gauge rails (see Fig. 10.3). For loads moved on their own wheels or on cars with flangeless wheels, the track consists of a channel laid flat on the floor (at one side) and a smooth wide strip (at the

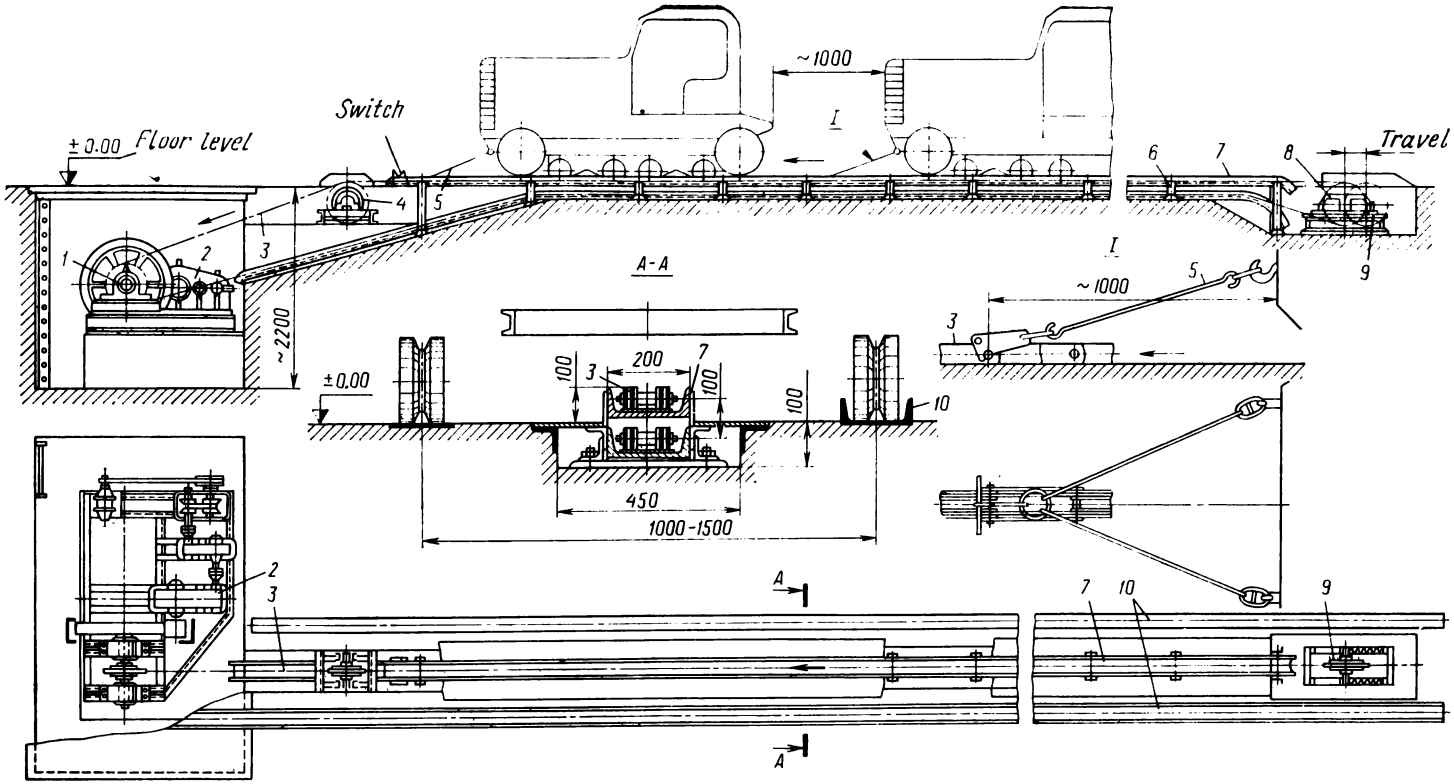


Fig. 10.1. Vertically closed load-towing carless conveyer for assembly of tractors

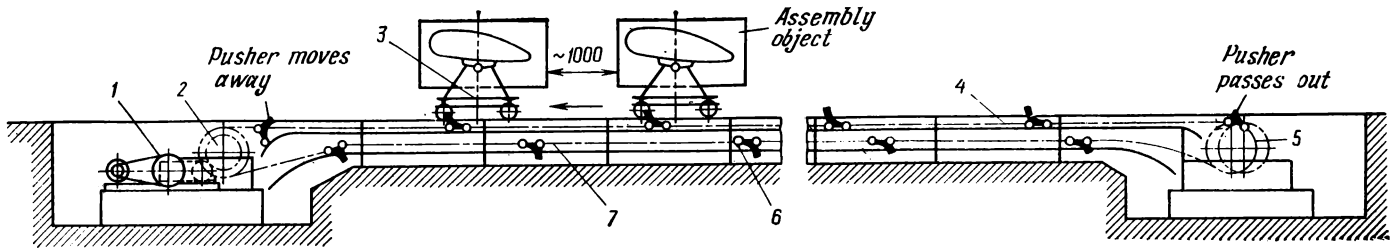


Fig. 10.2. Vertically closed load-towing car conveyer with a horizontal path

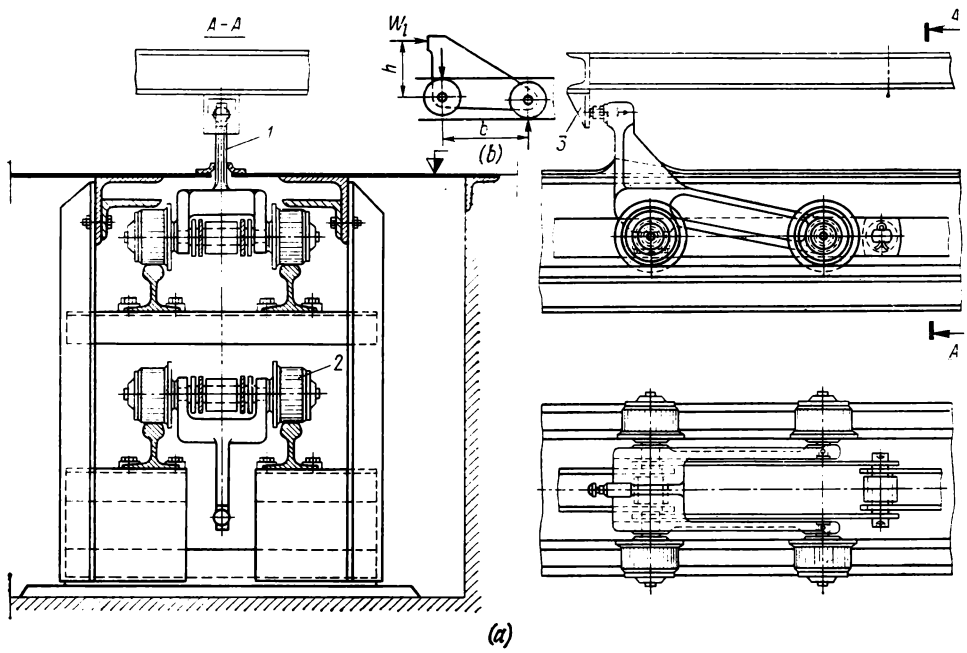


Fig. 10.3. The moving part of a load-towing car conveyer
(*a*) design; (*b*) calculation diagram

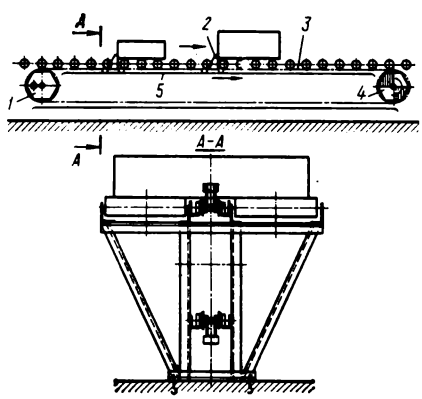


Fig. 10.4. Vertically closed tow conveyer with loads propelled on stationary rollers
1—take-up sprocket; *2*—pusher; *3*—hauling chain; *4*—drive sprocket; *5*—guide track

other), see Fig. 10.1. This design avoids seizure of wheels of the load or car. Railway cars are naturally moved on rails.

The drive is of the reducer type, sometimes with an additional chain or gear transmission. Variable-speed drives are employed when speed variation of a conveyer is essential. Damage of the conveyer is prevented by arranging a shear pin on one of the drive elements.

Steel-cable conveyers are driven by a reversible winch.

The take-up is of the screw or spring-screw type or, for cable conveyers, of the gravity type.

10.2.3. Conveyer Calculation

The main parameters of two conveyers can be determined by the formulae given in Ch. 2; the parameters of conveyers in process lines are determined by the formulae from Ch. 9. The length of tow conveyers may be up to 300 m and the speed 0.1-6 m/min in continuous motion or 6-12 m/min in intermittent motion.

The pull is found by successive calculation of resistances over the conveyer contour. Since the loads are moved on their own wheels, cars or a

roller table and the chain with pushers moves on wheels or slides on floor, the total resistance in the carrying run is determined as the sum of the resistances of the loads and chain. Besides, one should consider an additional resistance due to the off-centre or inclined application of the tractive force relative to the longitudinal axis of the chain. This force depends on the method of connection of the tractive element to the load or car. If a loaded car is propelled by cantilever pushers fastened on the chain (see Fig. 10.3*b*), the additional force

$$W_{ad} = W_l k_{ad} = W_l (2b/h) w_{ch} \quad (10.1)$$

where W_l is the resistance of the load N; $k_{ad} = (2b/h) w_{ch}$ is the coefficient of additional resistance; b is the spacing between the wheels; h is the arm length of the pusher (see Fig. 10.3*b*); and w_{ch} is the coefficient of the chain resistance.

With the chain sliding on a guide track, the coefficient of its resistance is equal to the coefficient of friction on lubricated steel and it can be taken that $w_{ch} = 0.25\text{-}0.35$. If the chain and cars move on wheels, their resistance coefficients are found by formula (2.23). For loads moved on their own wheels, it is advisable to determine the resistance coefficient experimentally, by measuring the force required to move the load by a dynamometer or, if this is impossible, to determine it approximately by formula (2.23).

For loads moved on rollers of a roller table, the resistance coefficient is found by formula (2.23).

10.3. Push-bar Conveyers

10.3.1. Design

A load-towing push-bar conveyer (Fig. 10.5) has one or two vertically closed combined tractive elements with pushers, a drive, and a take-up. The tractive element moves reciprocatedly and propels the loads from one position to another; the loads move on their own wheels or on one or two supporting cars.

The combined bar-and-chain tractive element in the carrying run of a conveyer comprises a number of successively connected beams or bars *3*, with sections of a chain (or cable) *2* fastened to their ends; the chains bend around a drive sprocket *1* and take-up sprocket (or sheave) *10*. The length of chain sections is 2-3 m greater than the length of working stroke of the conveyer. In the return run, the tractive element is the round steel tie *7* or steel cable of a diameter 6-8 mm. Bars *3* have support wheels *5* and guide pushers *6* by which the loads are propelled (on their own wheels or on cars) along rail tracks *9*.

On switching on of the drive motor *8*, the bars and chain sections begin to move and the pushers propel the loads forward through one stroke. An automatic switch then changes the direction of motor rotation and the bars return into the initial position.

Push-bar conveyers are less expensive compared with common vertically closed carrier chain conveyers, since large portions of the hauling chains are replaced by simple bars; they are simpler in manufacture and maintenance, especially at a substantial length and heavy loads.

Push-bar conveyers may have one or two strands of push bars (Fig. 10.5), with the two parallel strands being driven from a common drive. Two-strand push-bar conveyers are employed for transportation of very heavy and wide loads when the car track is greater than 3 m.

A special group is formed by push-bar conveyers with a rack-and-gear drive where the tractive element is made up of one or two rigid bars arranged along the entire length of the conveyer, and there are no flexible elements.

Push-bar conveyers are employed in assembling, manufacturing and repairing lines for heavy and large unit loads

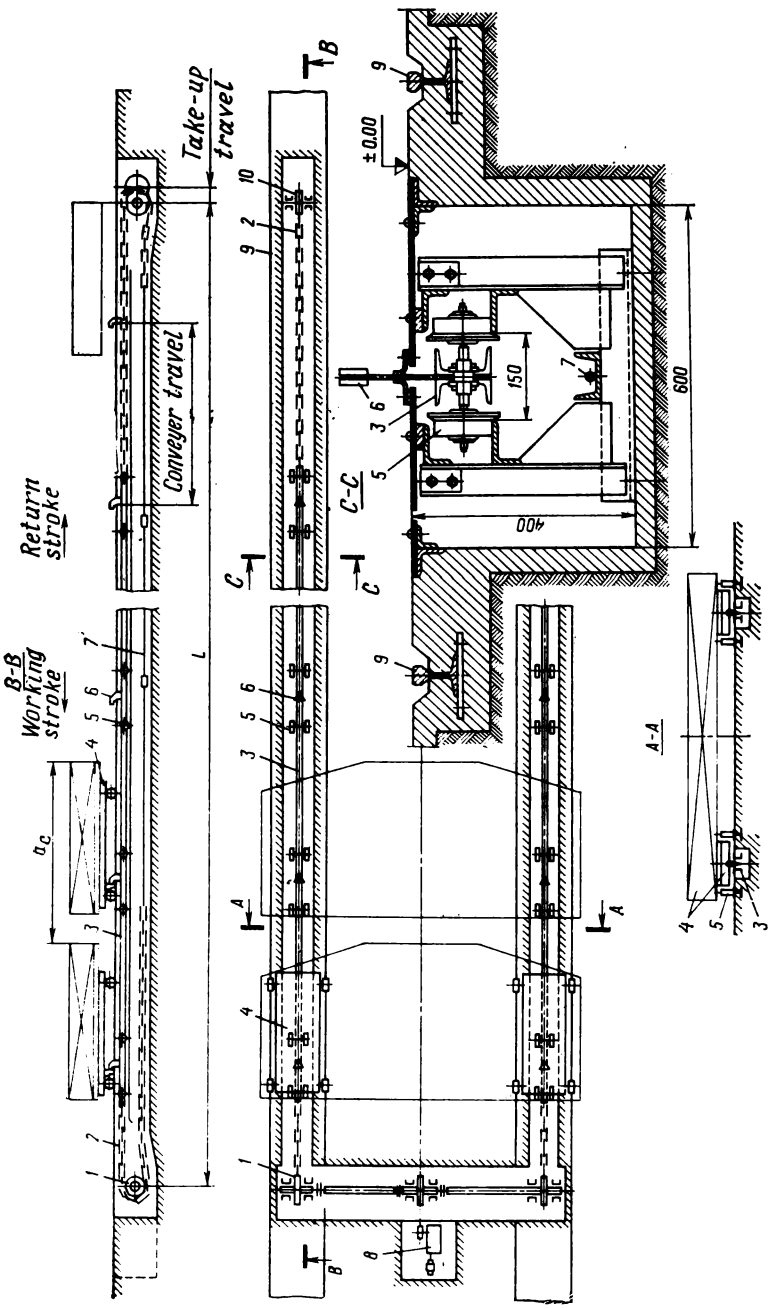


Fig. 10.5. Push-bar conveyer (USSR)

(railway cars, flat-cars, aeroplanes, reinforced concrete elements, etc.).

Push-bar conveyers have the following characteristics: push-bar speed 6-15 m/min; mass of loads 0.5-50 t and more; conveyer length 50-300 m; spacing between the two bars in two-strand push bar conveyers 3-8 m; and push bar stroke 5-30 m.

Return motion of push-bars is effected by reversing the drive motor. Special care should be taken that the pushers on both bars are in matched positions.

10.3.2. Conveyer Elements

The push bar is made from sections of I-beams, channels or prismatic rods. Support wheels and pushers are fastened to it. The bar moves on the wheels along the guide track on the conveyer structure and the pushers propel the loads. Pushers are fastened to the bar rigidly or on one-sided hinges at a certain spacing from one another which corresponds to the pitch of working positions on the conveyer. In some cases, the pushers are mounted with different pitch in order to diminish the tractive force at conveyer start-up; the loads are then pushed not at once, but one after another or in smaller groups, which decreases substantially the inertia forces.

The flexible portion of the tractive element is made from chains of any type or from steel cables. Steel cables are employed only in single-strand conveyers, since in two-strand conveyers they may be stretched differently and fail to ensure matched motion of the two bars.

The drive and take-up are of conventional type (see Sec. 10.2).

10.3.3. Conveyer Calculation

The force (N) to move the push bar in the working stroke is found by the formula:

$$W_{ws} \approx \xi g \left[w_c \left(\Sigma m_l + n m_c\right) + w_b m_b + \mu_0 m_{tr}\right] \quad (10.2)$$

and that in the return stroke,

$$W_{rs} \approx \xi g \left(w_b m_b + m_{tr} \mu_0\right) \quad (10.3)$$

where $\xi = 1.05$ is the coefficient of resistance of the chain on turning sprockets; w_c and w_b are the coefficients of resistance of a car and push bar; Σm_l is the total mass of all loads on the conveyer, kg; n is the number of cars; m_c and m_b are the masses of a car and push bar (or two bars in two-strand conveyers) together with wheels and pushers, kg; $\mu_0 = 0.25$ is the coefficient of sliding friction of the tractive element on guide tracks; and m_{tr} is the total mass of the tractive element (chain and push bars), kg.

A suitable standard size of chains and the power of drive motor are chosen for the force W_{ws} and the cross section of the return ties, by the force W_{rs}.

10.4. Horizontally Closed and Spatial-path Load-towing Car Conveyers with Automatic Addressing

10.4.1. Design

A horizontally closed and spatial-path on-floor car conveyer (Fig. 10.6) has a single hauling chain *1* running around a drive sprocket *5* and take-up sprocket *4*, and turning devices *2* and *3*, which may be arranged in a single or different horizontal planes at various levels. In vertical bends, the chain moves in guide tracks *6* as in overhead conveyers (see Ch. 8).

The hauling chain of the conveyer can be arranged on the floor (Figs. 10.6 and 10.7*a*) or beneath the floor in a narrow longitudinal pit (Fig. 10.7*b*). The chain is connected with load-carrying cars in the same manner as in overhead pusher conveyers (see Ch. 8). Pushers *11* are fastened rigidly (see Fig. 10.6) or by one-sided hinges to chain links. In the former case, load-carrying cars *7* are provided with a one-

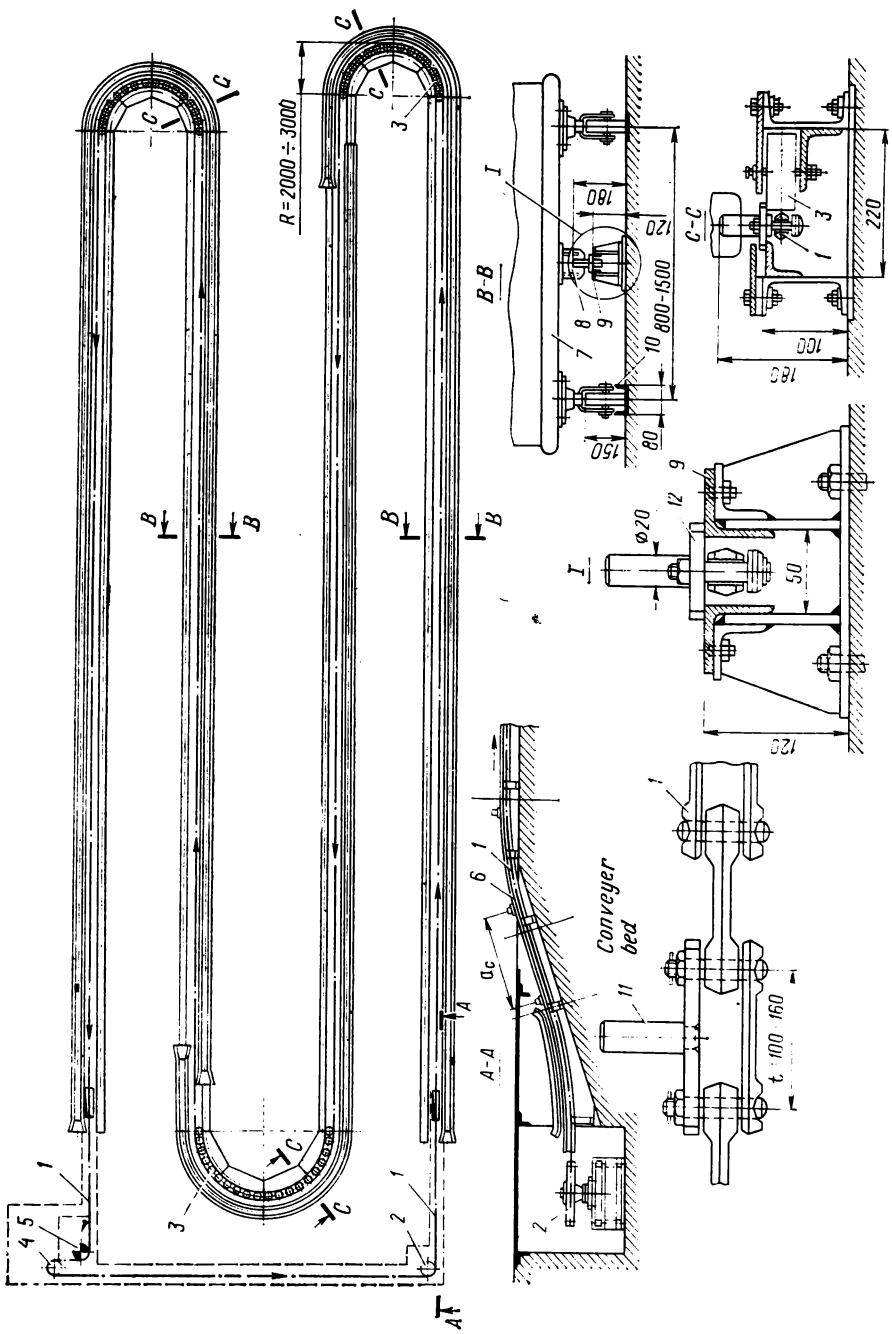


Fig. 10.6. Scheme and units of an on-floor car conveyer for automobile cabins

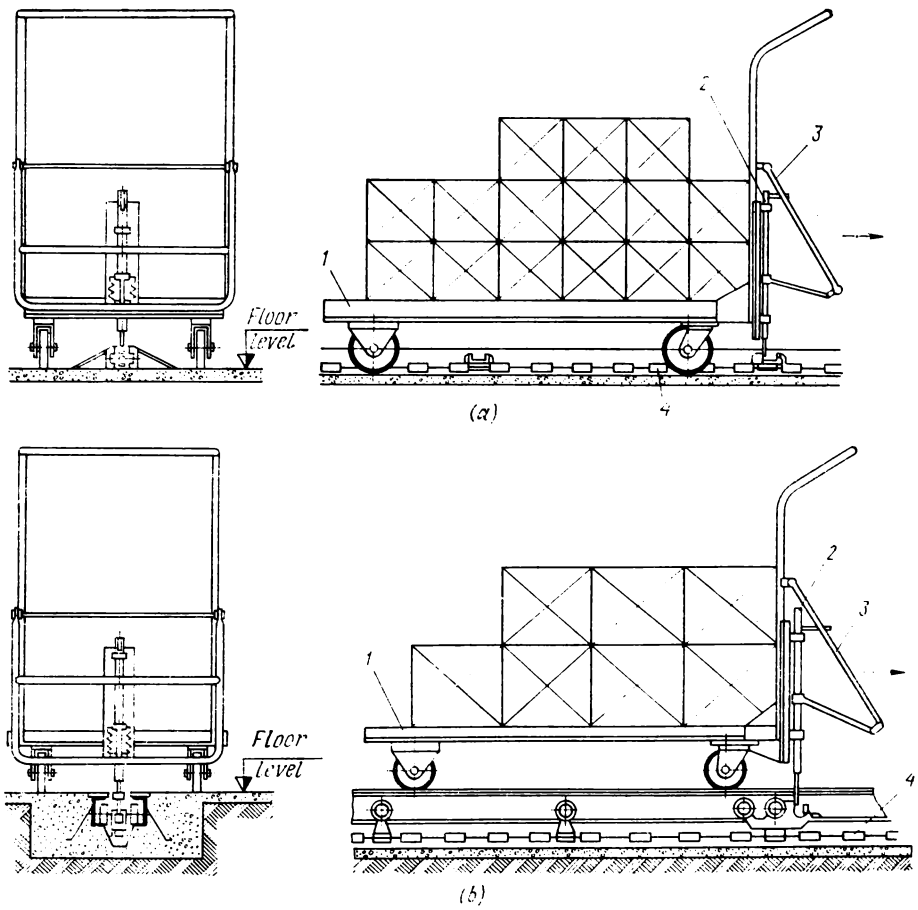


Fig. 10.7. Load-towing car conveyers with various arrangement of hauling chain (*a*) on the floor; (*b*) in the floor; *1*—car; *2*—leading pin; *3*—switch; *4*—hauling chain

sided hinged stop *8* or a vertically extending pin (Fig. 10.7*b*).

The hauling chain moves on guide tracks *9* (Fig. 10.6) and is supported by plates *12* or by wheeled cars (Figure 10.7). The cars move on carrying tracks *10* (Fig. 10.6) or more often, on the floor of the shop.

Tow conveyers with on-floor position of the tracks and chain (Figs. 10.6 and 10.7) are mainly employed in process lines (assembling, painting, drying, washing, etc.) with a simple horizontally closed path. Their principal drawback is that they block up the production area. On the other hand, they leave the floor intact (i.e. require no slot in the floor as is needed for in-floor tow conveyers) and the chains and tracks are easily accessible for inspection and repairs.

In-floor car conveyers (with the chain and tracks arranged under the floor, see Fig. 10.7*b*) have an intricate horizontally closed or spatial path with mechanical transfer of cars from one conveyer contour to another by an automatic addressing system (Figs. 10.8 and 10.9*a*). Automatically controlled switches can direct the cars onto sideways where the cars are catched by pushers of an auxiliary chain and pro-

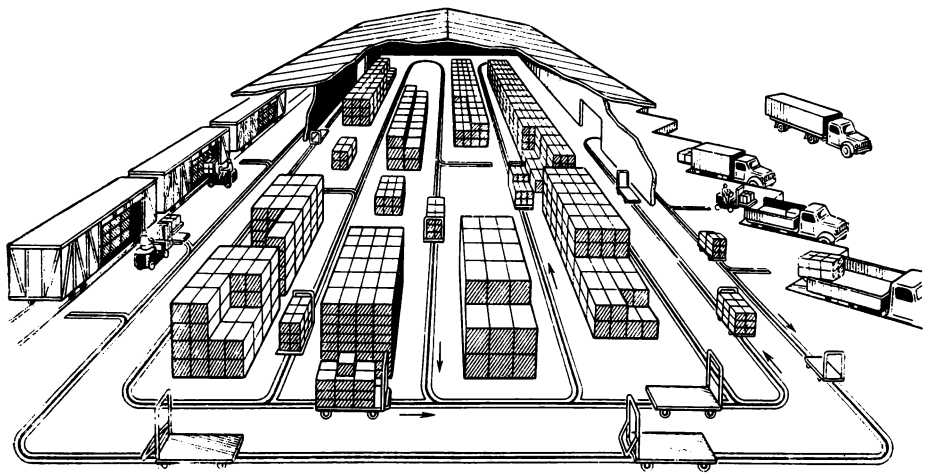


Fig. 10.8. Horizontally closed in-floor car conveyer in a railway store

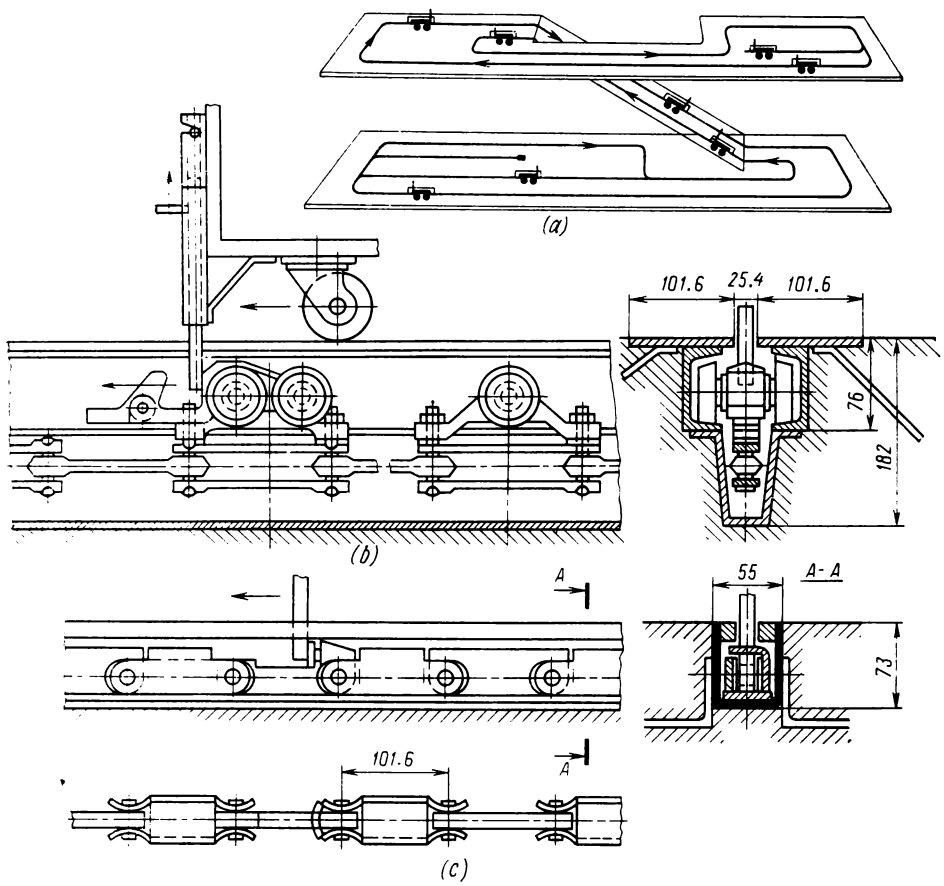


Fig. 10.9. In-floor car conveyers

(*a*) scheme of a conveyer spatial path; (*b*) conveyer with detachable chain (Webb Co., USA); (c) conveyer with laminated chain (Fromme Co., FRG)

pelled in a given direction. Cars are provided with address carriers of the automatic addressing system. Their address carriers interact with the address readers mounted along the conveyer path before the switches of the tracks, so that the cars are distributed automatically to the specified addresses. A car may be equipped with a non-controllable or controllable leading pin which is lifted and lowered by hand or by means of a mechanical arrester, so that a loaded car can be stopped in any point of the conveyer path.

An advantage of the in-floor car conveyers is that the shop floor remains free (since the chain is below the floor surface) for other on-floor vehicles and can be easily combined into complex transport-process systems with automatic addressing. Among their drawbacks are certain complications in inspection, maintenance, cleaning and repairs of the mechanisms arranged under the floor. The slot in the floor collects dust, dirt and foreign articles. It is difficult to install these conveyers on intermediate floors. In some conveyer designs, the slot is covered with rubber strips which protect the slot in the floor but let the pins of cars pass freely (see Fig. 10.5).

Notwithstanding their certain drawbacks, in-floor car conveyers are employed very widely in industry.

A typical example of mechanized transportation and handling of unit loads in a railway store with an extended system of in-floor car conveyers is illustrated in Fig. 10.8. Loads are delivered in railway cars and distributed in the store or supplied immediately to loading stages to be loaded onto automobiles. The loading-unloading operations on the stages are performed by truck loaders, while piling of loads in the store is carried out by piler cranes and other piling machines.

On-floor and in-floor car conveyers employ standard cars with the rated load-carrying capacity of 50, 125, 250, 500, 1000, 2000 or 3200 kg.

The tractive element is usually a detachable chain with a pitch of 80, 100 or 160 mm; cars move continuously at a rated speed of 0.3-47.5 m/min; the pit for the chain has a depth not more than 250 mm from the floor level (Fig. 10.9).

In foreign practice, conveyers of this type are known with cars of a capacity up to 20 t. The angle of incline of conveyer tracks usually does not exceed 15°. The length of single-drive conveyers is usually 50-500 m; multiple-drive conveyers may have a length up to 3 km.

On-floor car conveyers manufactured at the Lvov conveyer-making plant have cars of a load-carrying capacity of 250 kg, a double-hinged hauling chain with a pitch of 200 mm and tractive force up to 4.5 kN, and speed up to 25 m/min; load addressing is of the contactless type, with 15 addresses.

10.4.2. Conveyer Elements

The elements of equipment, control and automatic addressing of on-floor car conveyers have been largely standardized together with those of overhead conveyers which have been discussed in Ch. 8.

The hauling chain is detachable with a pitch of 80, 100 or 160 mm (Figure 10.9*b*), double-hinged with a pitch of 160 mm or 200 mm, or special laminated sleeveless (Fig. 10.9*c*).

The trolleys and hinged stops (Fig. 10.9*b*) are essentially of the same design as in overhead pusher conveyers.

The load-carrying cars have usually four or six wheels. Four-wheel cars have a rotatable (live) front axle and a rigidly fixed rear axle. In six-wheel cars, the front and rear axles are live and the mid axle is rigid, so that the car can pass freely and has a low resistance in conveyer turns.

Steel-rim wheels are employed rarely, since they have a high rolling resistance on uneven and dirty floors and cause intensive wear of the floor surface.

Steel wheels are effective when cars are moved on floors of steel plates, on guide tracks, or on floors fouled with oil, metal chips and chemicals. Wheels on pneumatic tyres move easily on any surface, but are depressed under load, which may change the level of the leading pin of a car and cause damage. Wheels with solid rubber rims are most popular and can move on any type of floor.

A load-carrying car has in its front portion a hinged safeguard which automatically disengages the leading pin from the chain should the car run onto an obstacle or another car.

The tracks are made from rolled or bent channel sections of a length of 3-6 m. Special inspection (repair) sections are provided in the conveyer path in intervals of 100-200 m, which are not filled with concrete and allow free access to the moving parts of conveyer for inspection, lubrication, cleaning, and repairs.

The drive is usually of the reducer type, as in overhead conveyers, but with the upper position of the drive sprocket. Angular and, more often, straight (caterpillar) drives are in use; multiple-drive conveyers are employed widely.

The take-up may be of the screw, spring-screw or hydraulic type.

The turning devices are roller sets, sprockets and sheaves. A turning device with a moving horizontally closed contour of wheel chain is employed in order to reduce the resistance and increase the service life of the hauling chain. The hauling chain is pressed against the wheels of the turning chain and entrains them. The wheels revolve on the chain and a guide strip. The wheel chains run around turning sprockets.

The automatic addressing and control system may be centralized or decentralized. Each of them has its preferable applications, depending on the particular conditions. It should be noted that the automatic addressing systems for load-towing car conveyers differ substantially from those for overhead conveyers due to different arrangement of the addressing elements on the cars and at switches, i.e. on and under the floor.

In a simple centralized system, addresses are recorded on a magnetic drum whose memory corresponds to the number of addressed operations in a conveyer. The drum rotates in synchronism with the conveyer drive, but at a substantially lower speed and simulates, as it were, the motion of the conveyer chain along the path.

A more complicated centralized addressing system contains a set of shift registers which simulate the motion of cars in the conveyer path. The distance between the starting point and the address of a load corresponds to a definite number of pulses supplied by a sensor which is connected mechanically with the conveyer chain. The logical portion of the electric addressing circuit is set up from standard ferrite-transistor modules and is quite compact and highly reliable.

Addressing operations are performed in the following manner. As a loaded car is introduced into the conveyer path, the operator presses a button on the control panel, which corresponds to a particular receiving address. When the car arrives to the destination point, a signal lamp is lighted on at the receiver and the switch is transferred to direct the loaded car from the conveyer to the receiver.

The main advantage of centralized systems is that there are no address carriers on cars and no readers before switches at destination points; all addressing elements are not distributed over the conveyer path, but concentrated in the control panel boxes, which ensures their better preservance and ease of maintenance. Among their drawbacks are complicated design and commutation and high cost.

Decentralized systems are simpler (especially with a low number of addresses) and exist in many contact-type and contactless versions which are eas-

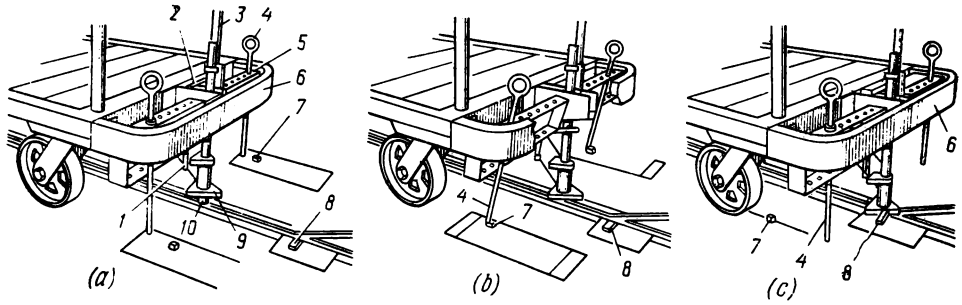


Fig. 10.10. Contact-type system for automatic addressing of cars (Webb Co., USA)

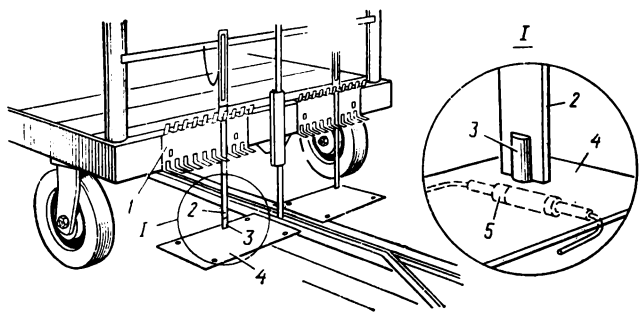


Fig. 10.11. Contactless magnetic automatic addressing system (Webb Co., USA)

ily adaptable to particular operating conditions. In the simplest design, a system has address carriers in the form of perforated plates arranged on the front wall of cars. The operator sets in one or two vertical pins into the plate perforations and the pins interact with the elements of the readers arranged along the conveyer path before switches. Address readers may be of contact type or contactless (photoelectric, magnetic or radioelectronic).

In a contact-type addressing system with a controlled (guiding) wedge (Fig. 10.10), the front portion *6* of a car has a two-sided tilting plate—address-carrying card *5* with perforations (usually up to 10 perforations at each side). Addressing pins *4* can be inserted into the perforations in various combinations. Two pins (one at the right-hand side and the other at the left-hand side) in various combinations can set up to 100 addresses. With the number of addresses up to 20, one pin is sufficient; it is inserted into the right-hand or left-hand portion of the card. The other pin is fixed in one of free perforations. Readers are provided in the conveyer path before switches to sideways. A reader has two address-reading stops *7*, which can be set up in various combinations across the path of addressing pins, and a deflecting stop *8*. When a car passes by a reader (Fig. 10.10*a*) and the setting of stops *7* and pins *8* is coincident, they deflect a plate *5* (Fig. 10.10*b*) and a lock *1*. The guiding wedge *9* with roller *10* is lowered along a leading bar *3* with handle *2*. In further motion, the car shifts the wedge *9* to contact with deflecting stop *8* and the latter moves the roller *10* into the slot of the sideway and the car is moved onto that way (Fig. 10.10*c*). The wedge is two-sided to ensure the transfer of cars to right- and left-hand sideways as required. The wedge is returned into the initial position by hand or automatically.

In a contactless magnetic addressing system (Fig. 10.11), a car has a two-sided address carrier *1* with slits into which one or two pins *2* with magnets *3* at their ends can be inserted. A reader *4* arranged under the floor has small-size sensors *5*. If the setting of the pins coincides with that of the sensors in

a reader, the latter sends a command to energize the switch of a sideway. Other versions of contactless addressing systems are also known.

Load-towing car conveyers are calculated as overhead pusher and overhead load-towing car conveyers (see Ch. 8).

10.5. Walking-beam Conveyers

10.5.1. Design

Walking-beam conveyers are employed for periodic (intermittent) transportation of unit loads between process operations in mechanical, assembling, heat-treating and other shops in various branches of industry and at prefabricated concrete plants. The loads are moved due to reciprocating horizontal and vertical ('walking') movements of a hinged frame on which they lie (Figure 10.12).

One of the first designs of walking-beam conveyer has been developed at the Giprostanok institute in the USSR for flow-lines for assembling of machine tools; various versions have later been designed for assembling of electric motors and other articles.

A walking-beam conveyer (Fig. 10.12) has a stationary frame *1* with guide rollers *2*, a moving frame *3* arranged inside the first frame, a number of lifting mechanisms *4* with support wheels *5* on which the moving frame is held, and a drive *6*.

The conveyer operates as follows. The load to be transported is laid in the first position on the stationary frame, while the moving frame is in the lowered position. To transfer the load into a next working position, the moving frame is raised by the lifting mechanisms roughly 20 mm above the level of the stationary frame and takes up the load; the drive is switched on and the moving frame moves forward by one step, i.e. through the distance between adjacent working positions. The frame is then lowered, the load sinks onto the stationary frame, and the moving frame is returned by the drive into the initial position.

With a fully loaded conveyer, the moving frame propels the loads at all

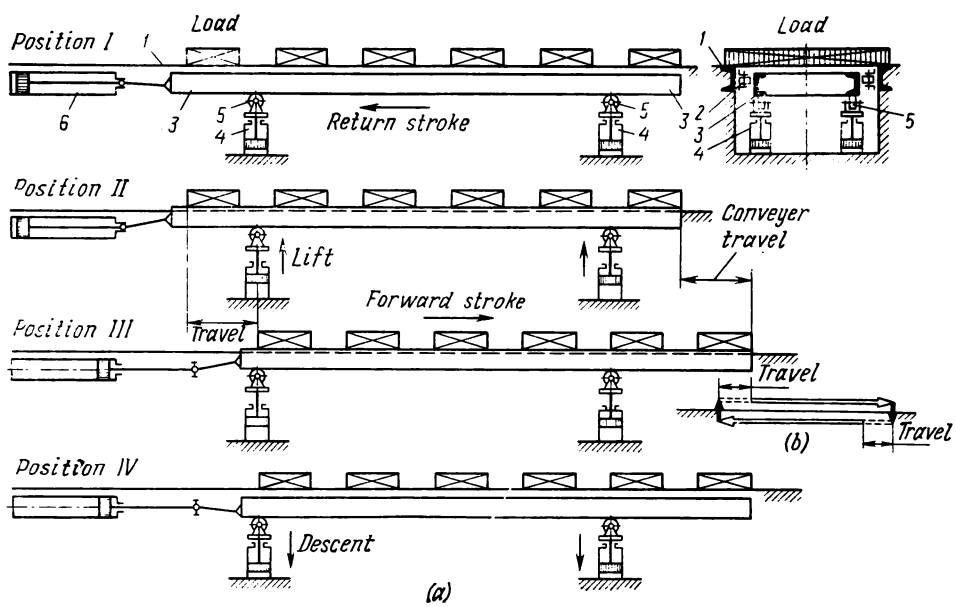


Fig. 10.12. Walking-beam conveyer (USSR)
(*a*) positions of frame during an operating cycle; (*b*) scheme of frame strokes

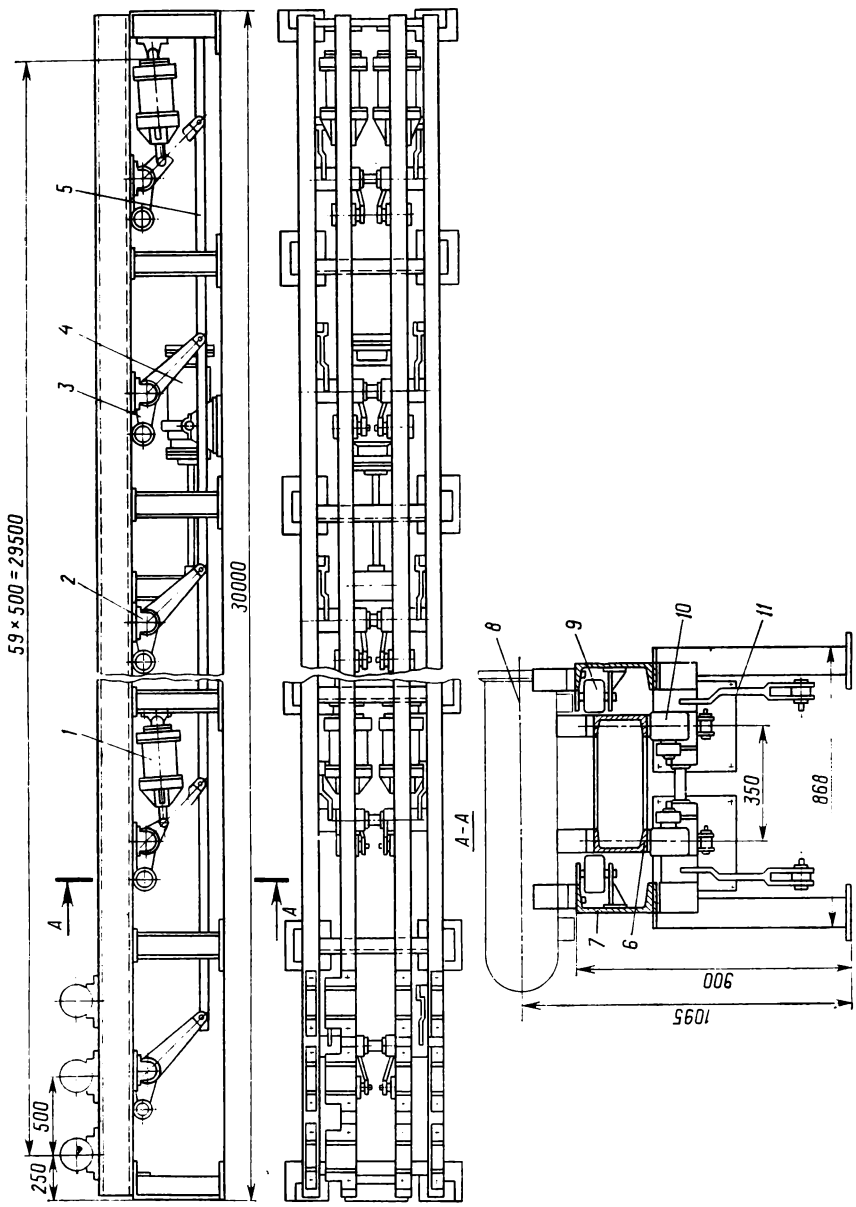


Fig. 10.13. Walking-beam conveyer with pneumatic drive (USSR)

working positions through one step in equal time intervals as required by the rate of the process. Thus, the entire cycle of a walking-beam conveyer is performed automatically in four successive motions of the moving frame: lifting, working stroke (forward motion), lowering, and return stroke (Fig. 10.12*b*).

The walking-beam conveyer with pneumatic drives for loads of a mass up to 200 kg, shown in Fig. 10.13, has a lifting pneumatic cylinder *1* with bearings *2* and lifting levers *3* and *11*, a pneumatic cylinder *4* for effecting the forward motion of loads *8*, ties *5*, moving frame *6*, stationary frame *7*, guide rollers *9*, and support rollers *10*. It operates essentially in the same way as described earlier (see Fig. 10.12*b*). The main characteristics of the conveyer are as follows: total load-carrying capacity 12 t; frame stroke 500 mm; air pressure in the pneumatic system 0.49 MPa; and total mass of conveyer 3.6 t.

The advantages of walking-beam conveyers include easy accessibility to loads, their steady position on the stationary frame, and relatively simple design as they have only a low number of moving elements and joints. Among their drawbacks are a limited configuration of the path (only a straight path is possible), impossibility of continuous motion, the necessity to make pits for the frame, lifters and drive, if the conveyer has to move load on the floor level (see Fig. 10.12), and substantial dynamic forces acting on the drive and supporting structures due to intermittent motions of the frame.

In complex conveying systems, walking-beam conveyers can be readily combined for interaction with overhead, cradle and roller conveyers and truck loaders for automatic overloading from one conveying machine onto another.

When it is essential to increase the number of load units on a conveyer at its given constant length, a two-stage walking-beam conveyer can be employed which represents a system of vertically closed conveyers (Fig. 10.14). It consists of two conventional walking-beam conveyers arranged one above the other and two hydraulic lifters to transfer loads from one conveyer onto the other. The upper walking-beam conveyer can be used, for instance, for assembling and filling of casting moulds, and the lower conveyer, for their cooling. The first lifter sinks the filled moulds from the upper onto the lower conveyer and the second lifter lifts cooled moulds onto the upper stage.

The system operates as follows. At start-up, the moving frames with moulds are lifted and both lifters are sunk (the first with a mould and the second, empty). The moving frame of the upper conveyer is then moved through a stroke leftwards, and that of the lower conveyer, is moved rightwards and takes up a filled mould from the first lifter and transfers a cooled mould to the second lifter. On attaining their extreme positions, the frames are lowered and at the same time both lifters are lifted (the first empty and the second with a mould). In their uppermost positions, the lifters actuate the drives of moving frames which are returned to the initial positions. In this operation, the first lifter carries a filled mould, while a cooled mould on the second lifter is taken up by a crane and transferred onto a knock-out grid.

The operating cycle of a two-stage walking-beam conveyer comprises four successive strokes of the mechanisms, with all their motions being interlocked and performed automatically.

A two-stage walking beam conveyer can save much area in a production shop and improve labour conditions in foundry shops, since mould cooling, in which harmful gases evolve, is carried out in an isolated underfloor. This type of conveyer can also be employed for transportation of loads in other processes, for instance, in assembling, painting and drying departments.

The number of load units on a walk-

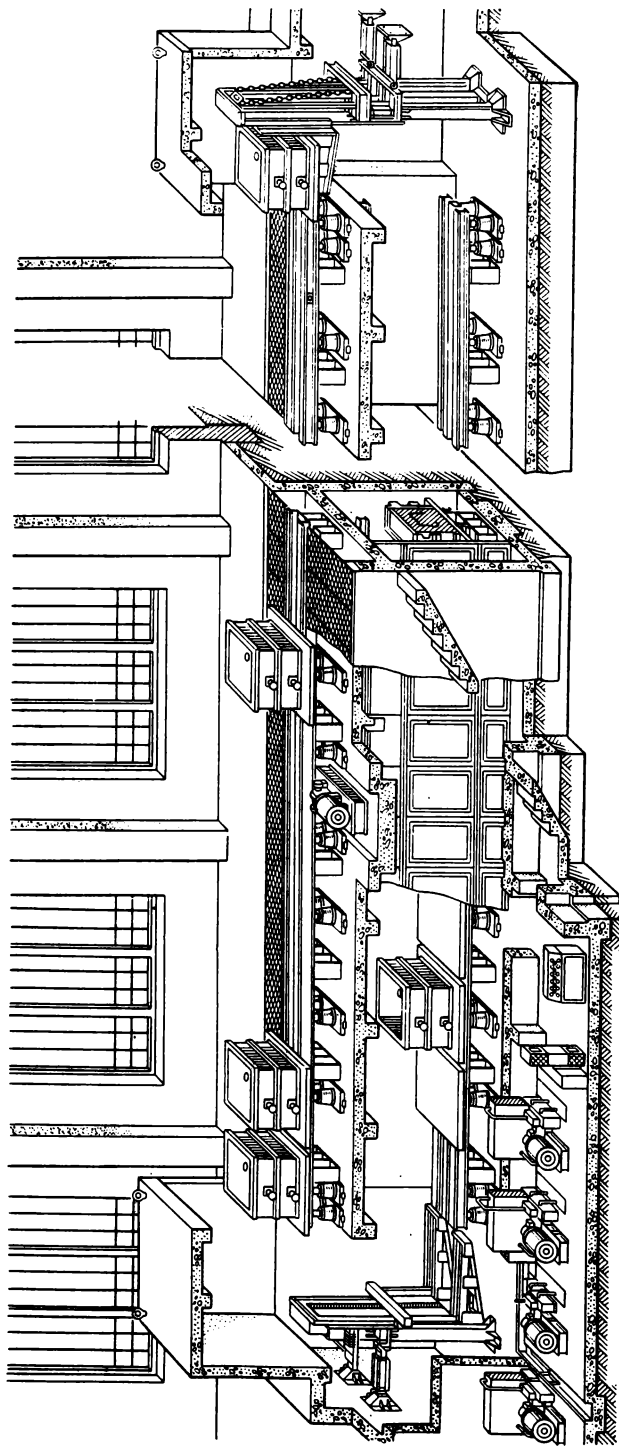


Fig. 10.14. Vertically closed two-stage walking-beam conveyer for casting moulds

ing-beam conveyer can be increased (if this is required in view of the restricted area of a shop) by arranging a number of walking-beam conveyers at right angles in the horizontal plane to form a closed conveying system. Loads can be transferred from one conveyer onto another by powered roller conveyers, by chain or pneumatic pushers. The transfer devices, however, decrease the reliability of such a system. A single horizontally closed car conveyer will be preferable in such cases, since it has no overloading stations and will be more reliable than a system of walking-beam conveyers.

The main characteristics of walking-beam conveyers are as follows: length 3-150 m; total load-carrying capacity 4-900 t; mass of loads 0.1-15 t; number of working positions 4-70; speed of moving frame 1-25 m/min; time of one cycle 12 s to 8 min; linear mass of conveyer 200-700 kg/m; width of stationary frame 0.4-3.5 m; and operation time 20-120 minutes or more.

10.5.2. Conveyer Elements

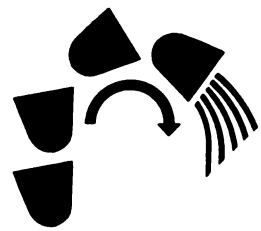
The frames are strong and rigid structures made from rolled sections (channels or I-beams); they are assembled from sections of a length of 3-6 m. The motion of the moving frame is guided by horizontal and vertical guide rollers. The top of the frame is covered by riffled steel sheets.

The drive may be electromechanical (rack-and-gear, crank or screw), pneumatic or hydraulic with pressures up to 6.5 MPa. Mechanical drives require a brake for exact fixation of a frame position.

The lifters may be pneumatic (for loads up to 1.5 t), eccentric or hydraulic (for loads above 1.5 t).

11

Bucket, Shelf and Cradle Elevators



11.1. Principal Types and Applications

Elevators are conveying machines for transportation of bulk and unit loads on vertical or steeply inclined (at an angle of 60-82° to the horizontal) paths and are accordingly divided into the vertical and inclined type. By the type of load-carrying element, they are classed into bucket, shelf (flight) and cradle elevators.

Bucket elevators are employed for conveying bulk materials—pulverized, granular and lumpy (such as cement, chemicals, sand, grain, flour, coal, peat, etc.) in the chemical, metallurgical and machine-building (in foundry shops) industries, in the manufacture of building materials and refractories, coal-dressing plants, granaries, food-processing plants, etc. They lift the loads from the initial to the final point, without intermediate loading or unloading.

Bucket elevators are divided into stationary and mobile (as elements of hoisting machines), transport and process elevators, for instance, desiccating elevators in which a wet material is freed from water when being lifted. A special type is an elevator with central loading and unloading and specially shaped buckets.

Cradle and shelf (flight) elevators are mainly used for lifting of unit loads: bags, bales, boxes, barrels and various machine parts in stores, depots, shops, and various industrial enterprises. Cradle elevators and some types of shelf elevators can move loads between floors of a building and allow intermediate loading and unloading.

A special group is formed by vertical cradle elevators (conveyers) for transportation of books between floors with automatic unloading, which are employed in large libraries.

Some kinds of bulk materials and small unit loads can be conveyed by double-belt conveyer-elevators in which the loads are pressed between two belts (see Ch. 4), and four-chain conveyer-elevators (see Ch. 7).

11.2. Bucket Elevators

11.2.1. Design and Main Types

A bucket elevator (Fig. 11.1) has a vertically closed tractive element *1* with load-carrying elements (buckets) *2* rigidly attached to it; the tractive element bends around an upper drive pulley *5* and lower take-up pulley *9* (or sprockets in chain elevators). The moving elements and the turning devices of an elevator are arranged in a clos-

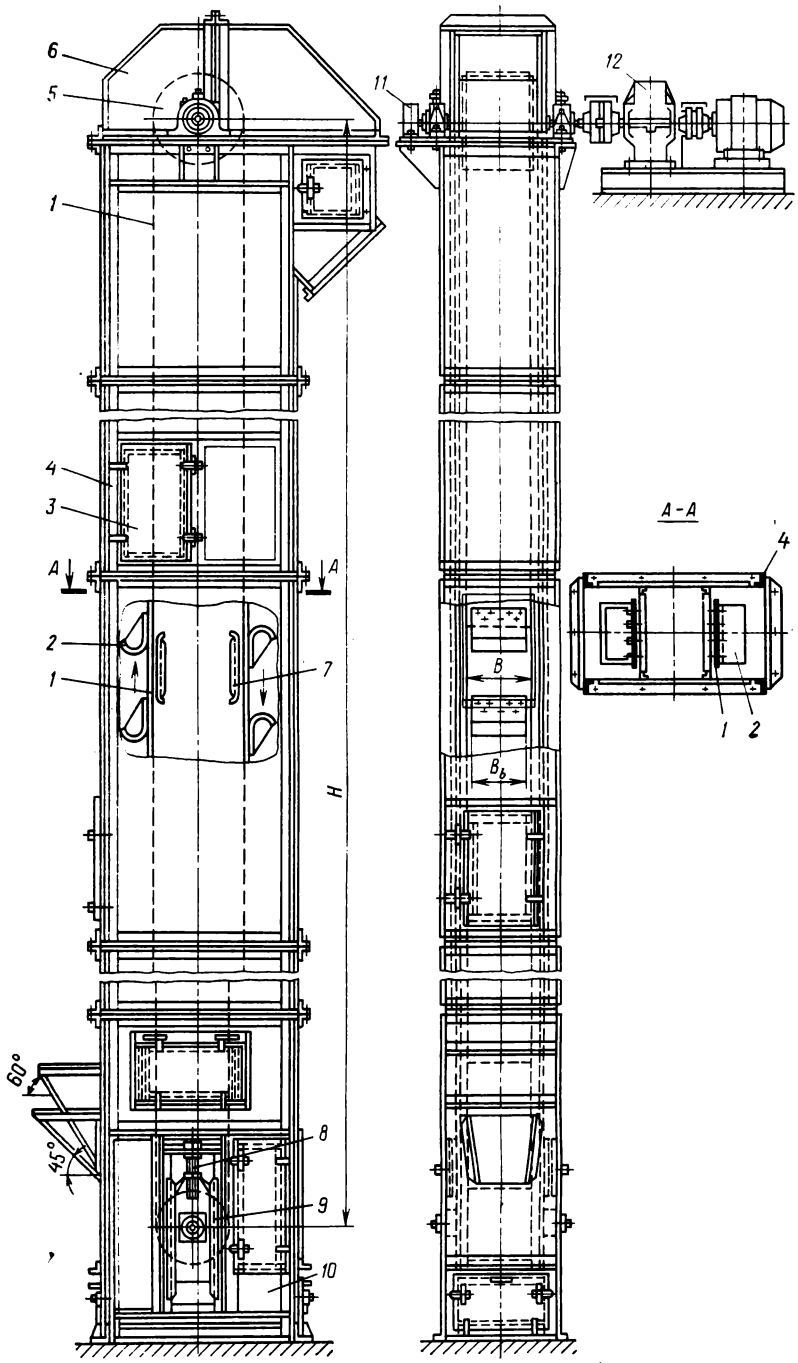


Fig. 11.1. Vertical belt bucket elevator

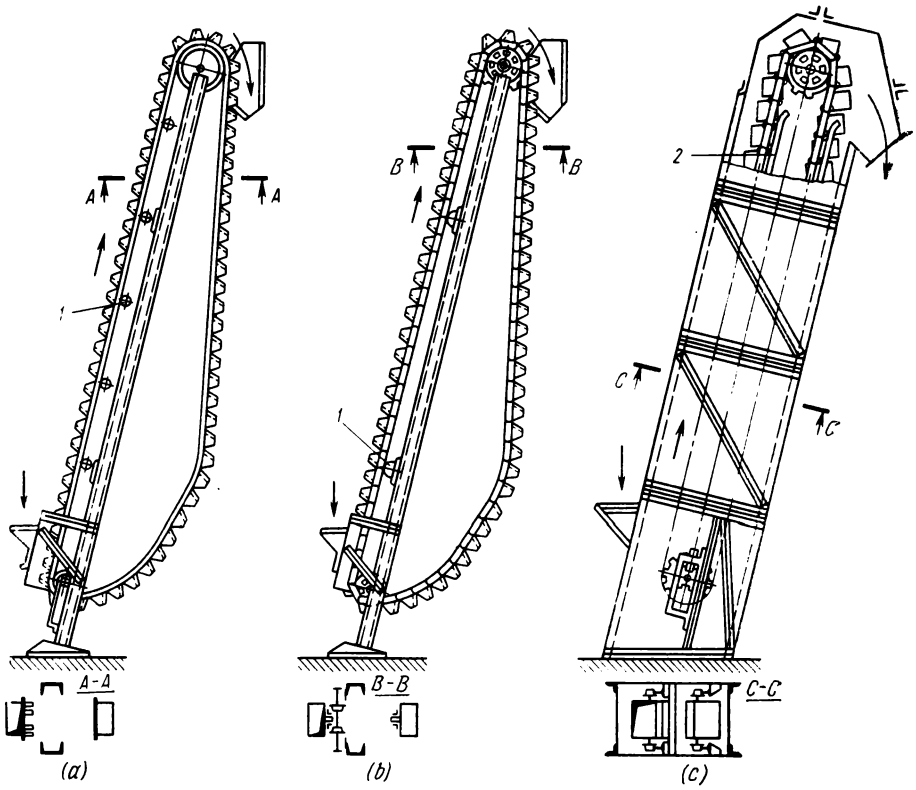


Fig. 11.2. Inclined elevators with closely placed buckets
(*a*) belt-type; (*b*) chain-type with freely hanging return run; (c) double-chain elevator with supported return run

ed metal casing which consists of a top portion ('head') *6*, middle sections *4*, and bottom portion ('boot') *10*. The tractive element with buckets is moved by a drive *12*, and the initial tension is set up by a take-up *8*. Bulk material is fed into the charging pipe (spout) at the elevator bottom, poured into buckets, lifted in them, and discharged at the upper pulley (sprocket) into a spout at the elevator top. The drive is provided with a stop *11* to prevent backward motion of the tractive element. The elevator casing has guide means *7* inside.

Among the advantages of bucket elevators are small cross-sectional dimensions, the possibility for lifting loads to an appreciable height (up to 60-75 m), and a wide range of throughput capacities (5-500 m^3/h and more). A drawback is that the material must be supplied in a uniform flow; on overloading, the buckets can break off from the belt or chains.

By the type of tractive element, there are belt elevators and chain elevators with one or two chains (designs with four parallel chains are also known). By direction of the path, they are divided into vertical (Fig. 11.1) and inclined (Fig. 11.2). In the latter, the return run can either sag freely (Fig. 11.2*a* and *b*) or be supported (Fig. 11.2*c*).

In inclined belt elevators, the carrying run of the belt moves on support rollers *1*; in chain elevators, chain links or (more often) chain wheels move in guide tracks *2* (Fig. 11.2*c*); in some

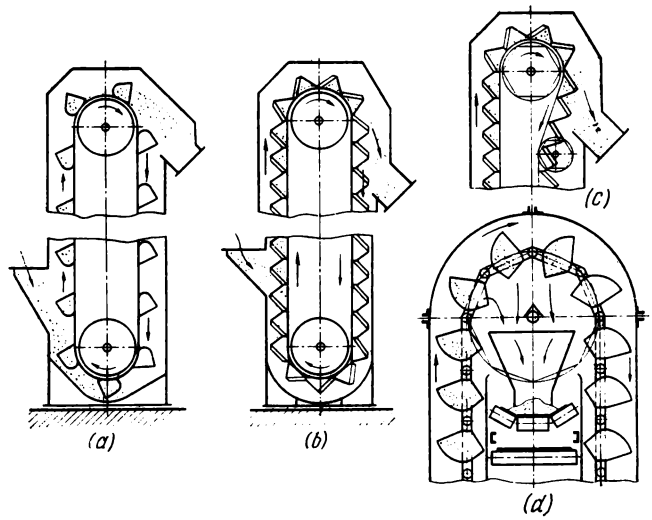


Fig. 11.3. Schemes of loading, unloading and bucket arrangement of an elevator
(*a*) loading by scooping, centrifugal unloading; (*b*) loading by pouring into buckets, directed gravity unloading; (*c*) free gravity unloading; (*d*) central unloading

designs of chain elevators, the carrying run of the chain moves on support rollers (Fig. 11.2*b*).

By the method of loading and unloading, bucket elevators are divided into high-speed ones which are unloaded mainly by centrifugal forces and low-speed elevators where the buckets are unloaded by gravity (see later in this chapter). An unloading spout can be arranged at a side of the elevator (Fig. 11.3*a-c*) or in the centre (Figure 11.3*d*). The latter method is applicable only in double-chain elevators and is employed only rarely.

The buckets of an elevator can be either spaced from one another (Fig. 11.3*a*) or placed closely (Fig. 11.3*b*). A bucket is always attached to the belt by its back wall. The arrangement of buckets (close or wide) is mainly determined by the characteristics of the conveyed load and predetermines the method of loading and unloading of buckets.

The main characteristics of general-purpose bucket elevators for bulk loads are given in Table 11.1. The speed of buckets may range from 0.4 to 2.5 m/s.

Special types of bucket elevators for

Table 11.1. Types of Vertical Bucket Elevators for Bulk Loads

Type	Name	Unloading method	Throughput capacity, m^3/h
BD	High-speed belt elevator with widely spaced deep buckets type D	Centrifugal	3.2-100
BS	High-speed belt elevator with widely spaced shallow buckets type S	Centrifugal	1.6-63
BV	Low-speed belt elevator with closely placed V-shaped buckets type V	Gravity	6.3-40
CD	High-speed chain elevator with widely spaced deep buckets type D	Centrifugal	3.2-100
CS	High-speed chain elevator with widely spaced shallow buckets type S	Centrifugal	1.6-63
CV	Low-speed chain elevator with closely placed V-shaped buckets type V	Gravity	6.3-40
CR	Low-speed chain elevator with closely placed round-bottom buckets type R	Gravity	40-320

vertical transport of grain and flour in grain mills, feed mills and granaries have maximum speeds up to 4 m/s, throughput capacity 5-500 t/h, and height up to 60 m.

11.2.2. Methods of Filling and Unloading of Buckets

Filling (loading) of buckets is performed by scooping the material on the bottom of an elevator (see Fig. 11.3*a*) or by pouring the material into buckets (Fig. 11.3*b*). In practice, buckets of an elevator are filled by both methods, one of them being prevailing. Scooping is applicable in belt and chain elevators with widely spaced buckets when conveying dry easily flowing, pulverized and fine-size lump materials (such as pulverized coal, milled peat, grain, cement, soil, sand, sawdust, crushed coal, rock phosphate meal, etc.) which offer only a small resistance and can be scooped at a higher speed of brackets (0.8-4 m/s).

Large-size lumpy and abrasive loads (gravel, ore, lumpy coal, etc.) are difficult to scoop by buckets; they offer a high resistance and can cause break-off of buckets or even of the tractive element. Large-size lumpy and abrasive materials are mostly poured directly into buckets. This method is only applicable in elevators with closely placed buckets and reduced speeds (not more than 1 m/s), since at higher speeds the buckets are filled poorly and throw off the material.

Buckets can be unloaded by centrifugal discharge (see Fig. 11.3*a*), free gravity discharge (Fig. 11.3*c* and *d*) or directed gravity discharge (Figure 11.3*b*). Centrifugal discharge is mainly due to the centrifugal forces which appear when buckets revolve over a pulley or sprocket. The transported material falls directly into the unloading spout of the elevator casing. For effective centrifugal unloading and in order to avoid fall-off of the material, it is very important to choose correctly the rotational speed of the pulley or sprocket and the position of the unloading spout relative to it.

Centrifugal discharge is employed in high-speed belt (and less frequently, chain) elevators with widely spaced buckets for conveying readily flowing pulverized, granular and fine-lumpy materials. The speed of buckets is usually 1 to 4 m/s. The spacing between buckets in high-speed elevators is selected so that the jet of discharged material could not strike the preceding bucket.

In free gravity discharge (see Figure 11.3*c*), the bucket is inclined and the material flows freely from it under the gravity force. This method of unloading is employed with poorly flowing moist, flaky and wet materials (such as pulverized coal, chalk, various chemicals, wet ash or sawdust).

In vertical elevators, free gravity discharge is performed by snubbing the return run on additional guide sprockets, roller sets (rarely) or guide bars (in two-chain elevators with buckets fastened between the chains). In single-chain elevators, this deflection (snubbing) is only possible if the centrally arranged chain carries buckets at its sides, while its internal and external surfaces are free to engage the snubbing sprockets.

In inclined elevators, free gravity discharge is ensured by the inclined position of the elevator. For that reason, vertical elevators are sometimes made with their top portion being inclined, which however increases resistances and results in quicker wear of the chain and guide rails.

The principle of free gravity discharge is also employed in double-chain elevators with central unloading (between the chain runs, see Fig. 11.3*d*); they have a reduced speed of buckets (0.6-0.8 m/s).

Directed gravity discharge (see Figure 11.3*b*) is typical of vertical and inclined elevators (either belt or chain type) with closely placed (continuous) buckets. During revolving on the top

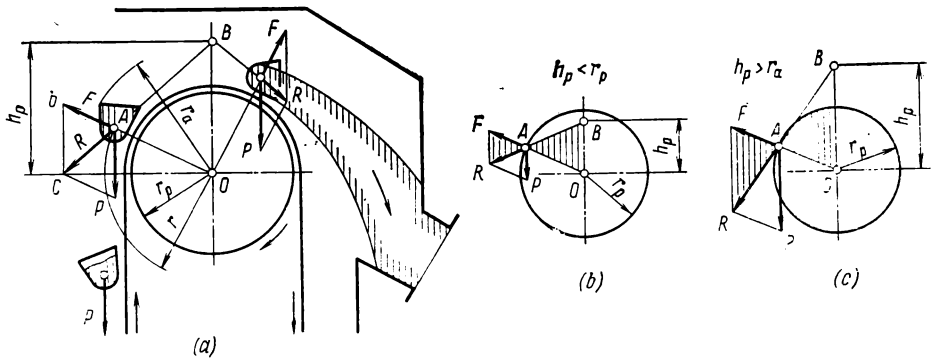


Fig. 11.4. Diagrams to calculate the pole distance

pulley or sprocket, the material pours from a bucket onto the back surface of a previous bucket and is directed by this and by the side walls of buckets into the unloading spout of the elevator. This method is employed in low-speed elevators (with bucket speed of 0.4-0.8 m/s) for conveying lumpy, heavy (gravel, ore, slag, large-size coal, etc.) and brittle materials (peat, charcoal, coke, etc.) which might deteriorate on being crushed.

Central loading and unloading of buckets is employed in elevators manufactured by Schenck Co., the FRG.

Principal criteria of bucket unloading. In the ascending run of a vertical elevator, the load is in a straight uniform motion and acted upon by a constant gravity force $P = mg$ (Fig. 11.4*a*). As a bucket begins to move around the top pulley or sprocket, an additional centrifugal force appears

$$F = mv_0^2/r \qquad (11.1)$$

where m is the mass of load in the bucket, kg; v_0 is the velocity of the centre of gravity of the bucket load, m/s; and r is the radius of rotation, i.e. the distance from the centre of gravity A of the load to the centre of pulley O, m.

As a bucket revolves on the pulley, the resultant R of the forces P and F varies in magnitude and direction. If, however, the resultant force vector is prolonged to intersect the vertical line passing through the pulley centre, it turns out that at any position of the bucket, vector R intersects the vertical in one and the same point B called the pole. The distance h_p from point B to the pulley centre O is called the pole distance.

From the similarity of the triangles ABO and Abc we have

$$h_p/r = P/F = mg/(mv_0^2/r)$$

whence the pole distance

$$h_p = gr^2/v_0^2 \qquad (11.2)$$

Substituting $v_0 = \pi rn/30$, we obtain

$$h_p = gr^2 \times 30^2/(\pi^2 r^2 n^2) = 895/n^2 \qquad (11.3)$$

where n is the rotational speed of the pulley, rpm.

Therefore, the pole distance h_p depends solely on the rotational speed of the pulley (or sprocket). At higher speeds, the pole distance decreases, while the centrifugal component increases as compared with the gravity force. The same conclusion has been drawn upon analysis of unloading from the pulley of a belt conveyer (see Ch. 4).

At lower speeds of the pulley, the effect of the gravity force on unloading becomes stronger and h_p increases. As has been demonstrated by P.S. Kozmin when h_p is not larger than r_p, i.e. the pole is inside the pulley circle (Figs. 11.4*b* and 11.5*a*), the centrifugal

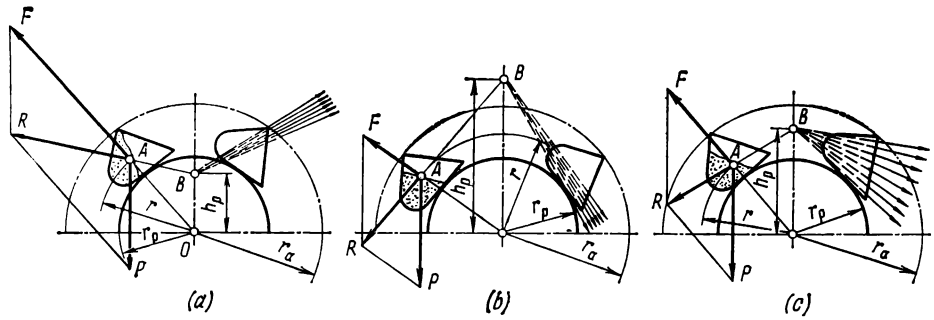


Fig. 11.5. Forces acting during bucket unloading
(a) centrifugal unloading; (b) gravity unloading; (c) combined unloading

force is much higher than the gravity force; particles of the material in a bucket are displaced to the external (front) wall of the bucket and the latter is unloaded centrifugally. With h_p greater than r_p, i.e. with the pole being beyond the circle described through the front edges of buckets (Figs. 11.4*c* and 11.5*b*). the force of gravity is large compared with the centrifugal component and the buckets are unloaded by gravity over their back wall (closest to the pulley). With $r_p < h_p \leqslant r_a$, unloading of a combined type (centrifugal and by gravity) takes place (Fig. 11.5*c*).

Therefore, the method of bucket unloading is determined by the ratio between the pole distance and pulley radius

$$A = h_p/r_p \qquad (11.4)$$

From formula (11.2) we have

$$A = h_p/r_b = gr^2/(v_0^2 r_p) = gr_p/v^2 \qquad (11.5)$$

where v is the circumferential speed of the pulley, m/s.

Thus, the kind of unloading of elevator buckets is determined not by the magnitude of their velocity, but by the ratio of that velocity to the pulley diameter. It is possible to perform centrifugal unloading at a relatively low bucket speed by using a pulley of a small diameter and, on the other hand, to provide gravity unloading at a high speed by increasing the diameter of pulley.

N.K. Fadeev recommends the following ratios between the diameter of pulleys (or of the base circle of sprockets), D_p in m, and the speed v of the tractive element (m/s) for various methods of unloading:

for high-speed elevators with centrifugal unloading

$$A \leqslant 1; \quad D_p = 2Av^2/g \leqslant 0.204v^2 \qquad (11.6)$$

for high-speed elevators with combined (centrifugal and gravity) unloading

$$A = 1\text{-}1.4; \quad D_p = (0.205\text{-}0.286)v^2 \qquad (11.7)$$

for moderate-speed elevators with combined unloading over the back wall of buckets

$$A = 1.5\text{-}3; \quad D_p = (0.306\text{-}0.612)v^2 \qquad (11.8)$$

and for low-speed elevators with gravity unloading

$$A > 3; \quad D_p \geqslant 0.6v^2 \qquad (11.9)$$

The trajectory of particles discharged from a bucket. As a bucket moves around the top pulley, a particle of load in it is acted upon by the gravity force, centrifugal force, and inertia force due to the Coriolis acceleration and relative acceleration of sliding of the particle on bucket wall. The path

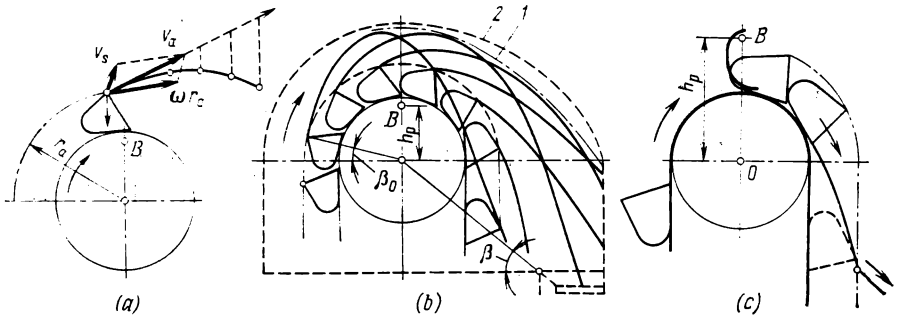


Fig. 11.6. Diagrams to determine the paths of ejection of material from buckets
(*a*) diagram of velocity components of material particles; (*b*) paths of particles in centrifugal unloading with consecutive positions of buckets (*1*—deviation line; *2*—profile of head casing of elevator); (c) in gravity unloading

and velocity v_s of a sliding particle can be determined by solving the equation of motion of the particle. The absolute velocity of particle, v_a, is found as the geometrical sum of the circumferential velocity $v = \omega r_a$ and the velocity of sliding v_s (see Fig. 11.6*a*). When thrown off from a bucket, a particle will move along a parabola. The start of free motion of particles depends on the method of unloading. With centrifugal discharge (Fig. 11.6*b*), the load begins to move in the bucket as the latter has rotated through an angle $\beta_0 = 15\text{-}30°$. Typical trajectories of particles in centrifugal and gravity unloading are illustrated in Fig. 11.6.

The profile of the head casing of an elevator is given the shape according to the trajectories of material particles in unloading. Its configuration should be such that all ejected particles are directed into the unloading spout and do not strike the casing walls, which might cause excessive crushing and dusting. The position of the unloading spout is determined by the unloading angle β. For elevators with centrifugal or combined unloading, it may be taken that $\beta = 30\text{-}45°$. The unloading spout should be arranged so as to avoid fall-through of the material onto the return run of the elevator.

11.2.3. Elevator Elements

Buckets. The principal parameters of elevator buckets are their geometrical dimensions (width B, length L, and height H) and volume. The type of buckets is chosen according to the properties of the material to be transported and the methods of loading and unloading. Vertical elevators may have buckets of the following main types: deep (type D in Fig. 11.7*a*), shallow (type S in Fig. 11.7*b*), V-shaped with guiding side walls (type V in Fig. 11.7*c*), and those with guiding side walls and round bottom (type R in Fig. 11.7*d*). Trapezoidal buckets of increased volume and other special types of buckets are also known. Inclined elevators preferably employ buckets with guiding side walls (V-shaped or with round bottoms) and trapezoidal buckets of increased volume.

In deep buckets (Fig. 11.7*a*), the top side edges have only a small angle of incline from the horizontal; they have an appreciable depth and are employed with dry easily flowing pulverized, granular and small-size lumpy materials (such as grain, sand, soil, fine coal, etc.). In two-chain conveyers with gravity unloading, with the buckets fastened by their side walls to the two chains (Fig. 11.7*a*, version *3*),

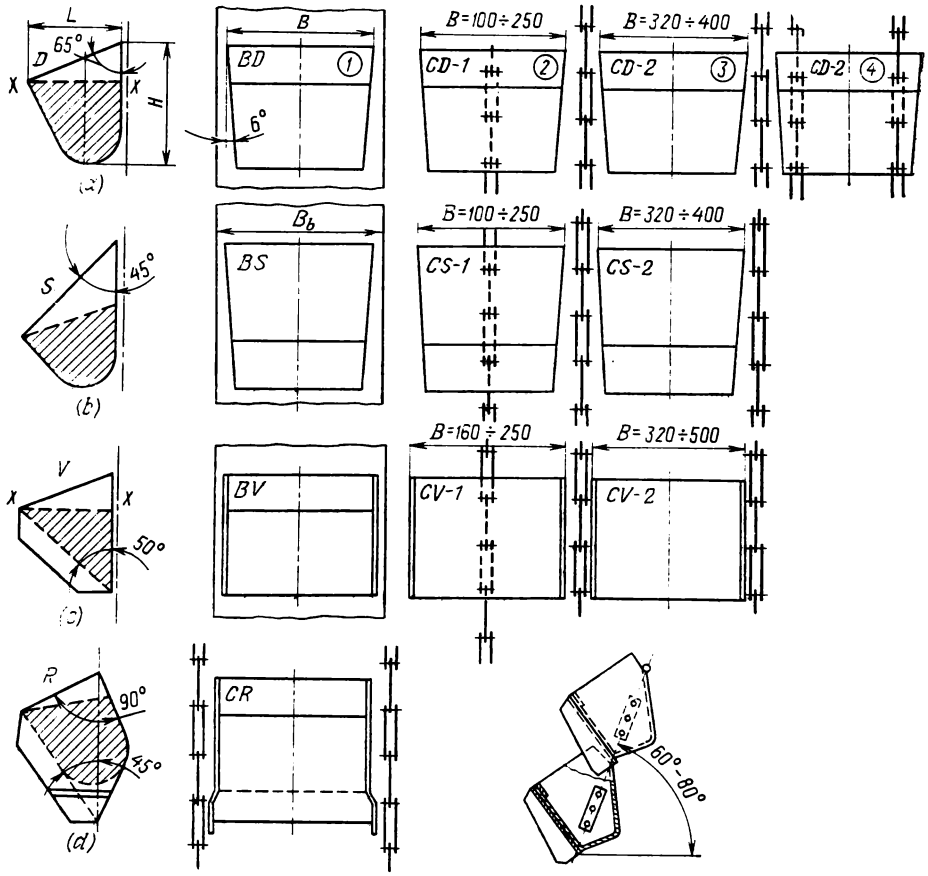


Fig. 11.7. Shapes of buckets and their attachment to the tractive element
1-4—versions of attachment of deep buckets to belt or chain

deep buckets can carry some bulk materials of a poor flowability (for instance, soot, powder lime, etc.).

Shallow buckets (Fig. 11.7*b*) have steeply inclined top side edges and a low depth, because of which they are emptied more easily in unloading. This makes them especially suitable for transporting of wet and poorly flowing pulverized, granular and fine-size lumpy materials.

Deep and shallow buckets mostly have a rounded-off (cylindrical) bottom for better emptying and decreasing the sticking of material particles to the bottom.

In some designs of bucket elevators with widely spaced buckets, the problem of easy emptying has been solved by employing bottomless buckets together with common ones. Two or three buckets on a conveyer have no bottoms, then follows a common bucket with bottom, and so on. During lifting, the material in bottomless buckets is retained by forces of internal friction of material particles and is held up by the material in the common bucket beneath them, but is discharged from them more easily in unloading.

Buckets of the deep and shallow types are employed only in elevators with widely spaced buckets. They are made from steel sheets 1-6 mm thick by welding or stamping or sometimes cast from pig iron. Buckets made of

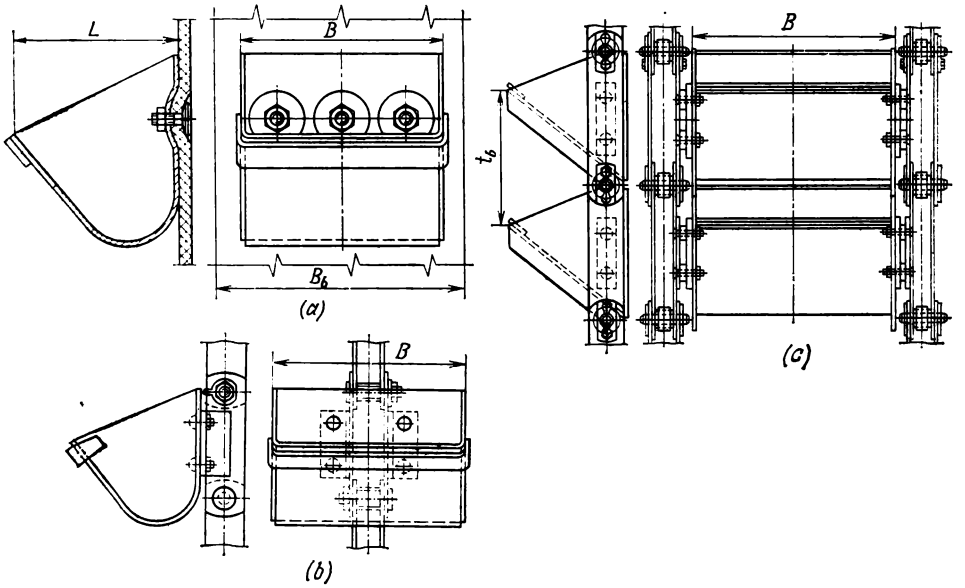


Fig. 11.8. Bucket attachments
(a) to belt; (b) to a single chain; (c) to two chains

plastics (fibrous plastics, glass fibre) and rubber are also known. Quick wear of the front (scooping) edge of buckets is prevented by welded-to or riveted plates of hard steel.

V-shaped buckets with guiding side walls (Fig. 11.7*c*) are employed in low-speed chain elevators for conveying diverse bulk loads: pulverized, granular and lumpy. All types of buckets with guiding side walls are arranged closely on a belt or chains. In desiccating elevators, buckets have perforated bottoms to drain water.

The main characteristics of standard buckets are given in Table 11.2.

In foreign practice, buckets of a width up to 1600 mm are known.

The tractive element. The tractive element in bucket elevators is made up of a belt or one or two chains. The belt may be of a standard rubber-fabric or rubber-cable type as in belt conveyers (see Ch. 4). Buckets are fastened to the belt by special flat-head bolts (Fig. 11.8*a*), so that the back surface of the belt, which runs around a pulley, remains smooth; the holes for the bolts in the back walls of buckets are recessed. Various washers or spacers can be provided to prevent accumulation of material in the gaps between buckets and belt (Fig. 11.9); they are fastened to belts or buckets by vulcanization.

The width of the belt should be 25-150 mm greater than that of buckets.

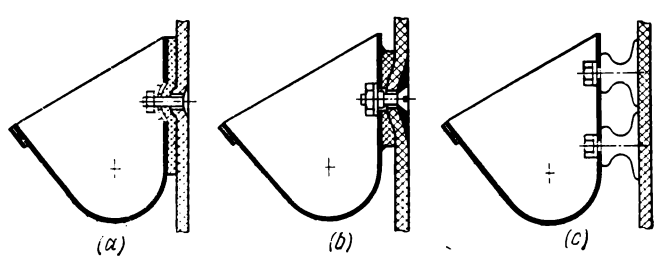


Fig. 11.9. Bucket attachments (Beumer Co., FRG)

Table 11.2. Main Chara eristics of Buckets (see Fig. 11.8)

Bucket width B, mm	Belt width B_b (for belt elevators), mm		Pitch of widely spaced buckets, t_b, mm	Deep buckets, type D		Shallow buckets, type S		Pitch of closely placed buckets, mm	Buckets with side guides			
									V-shaped, type V		round-bottom, type R	
	first row	second row		i_0, l	i_0/t_b, l/m	i_0, l	i_0/t_b, l/m		i_0, l	i_0/t_b, l/m	i_0, l	i_0/t_b, l/m
100	125		200	0.2	1	0.1	0.5	—	—	—	—	—
125	160	150	220	0.4	1.3	0.2	0.66	—	—	—	—	—
160	200	—	320	0.6	2	0.35	1.17	160	0.65	4.06	—	—
200	250	—	400	1.3	3.24	0.75	1.87	200	1.3	6.5		
250	300	315	400	2.0	5	1.4	3.5	200	2	10		
320	370	400	500	4.0	8	2.7	5.4	250	4	16	6.4	25.6
400	450	500	500	6.3	12.6	4.2	8.4	320	7.8	24.4	14	43.7
500	550	600	630	12	19	6.8	10.8	400			28	70
650	700	—	630	16.8	26.6	11.5	18.2	500	—	—	60	120
800	—	—	—	—	—	—	—	630			118	187
1000	—	—	—	—	—	—	—	630			148	235

The number of plies in the belt is determined by pull calculation, after which a standard belt of the required strength can be selected. Elevator belts are calculated in the same way as those of belt conveyers (see Ch. 4); it should only be considered that holes for bucket fastening diminish their strength.

The weakening effect of bucket holes can be neglected if the belt ends are joined by metal members (the strength coefficient of such a joint $K_j = 0.4\text{-}0.5$, see Ch. 4).

The belt must contain at least four plies to have a strength sufficient to hold the buckets. Rubber-cable belts are employed in elevators of a large height and with wide buckets. Articulated wire belts are also known to be used in elevators for hot loads.

Elevator chains are mostly laminated chains (sleeve-type, roller or wheel, the last ones preferably for inclined elevators with the supported return run) with a pitch of 100-630 mm or welded round-link chains from steel rods 16-28 mm in diameter with heat-treated links. The choice of chain type (either laminated or round-link) is mainly dictated by the characteristics of the load. When carrying pulverized and granular abrasive materials or those causing metal corrosion, joints of laminated chains can be blocked up with material particles and lose flexibility. This is prevented by increasing the gaps between the pins and sleeves up to 0.4-0.6 mm and by increasing their surface hardness to 58-62 HRC by carbonitriding.

The materials indicated do not block up the open joints of round-link chains (since particles are not retained in them), but can cause noticeable wear. Wear of round-link chains can be diminished by their heat treatment to a depth of 2 mm to 55-60 HRC.

Elevators with heat-treated round-link chains and centrifugal or combined unloading have found wide application for transportation of pulverized materials.

Buckets are fastened to chains by means of angle or shaped sections with bolts or rivets. With the bucket width up to 250 mm, a single hauling chain is used, which is fastened in the middle of the back wall of buckets (Fig. 11.8*b*). Wider buckets (320 mm or more) re-

quire two chains and are fastened to them by the back or side walls (Fig. 11.8*c*).

A belt or chain for an elevator is selected, depending on the throughput capacity, lifting height, and characteristics of load. Rubber-fabric belts can be employed at higher speeds than chains and are less subject to wear by abrasive materials, but have a lower pull and lower strength of bucket fastenings. For that reason, rubber belts are mainly employed in high-speed elevators for dusty, pulverized and fine-size lumpy materials of low or moderate density which have a low resistance for scooping. Chains are preferably used in elevators of large throughput capacity and substantial height for transportation of heavy-lump and hot materials which cannot be transported by rubber-belt elevators.

Abrasive materials are transported where possible by belt elevators, since chains would be worn off quickly. Rubber-cable belts make it possible to increase substantially the tractive force and the height of elevator. Both belt and chain elevators may be with closely placed or widely spaced buckets.

The drive. Elevators have a reducer-type drive arranged in the elevator head. Geared motors are employed in low-power elevators (up to 10 kW). The application of straight intermediate drives may be effective in chain elevators of a substantial height.

The diameter of the drive pulley, D_p, for belt elevators is determined depending on the method of bucket unloading [see formulae (11.6)-(11.9)] and the number of plies i in the belt [usually D_p = (125-150) i, mm], after which the closest pulley size is chosen from the standard series: 250, 320, 400, 500, 630, 800, and 1000 mm. Rubber-lined pulleys are usually employed in order to increase the coefficient of friction with the belt.

The drive sprockets for laminated-chain elevators should have 5-20 teeth. Elevators with round-link chains have a friction-type drive with drive sheaves having a smooth shaped rim or with cogged-teeth sprockets; their diameter is chosen from the standard series of diameters of sheaves and sprockets. The shafts of drive pulleys and sprockets are mounted in self-adjusting rolling bearings. Spontaneous back motion of the tractive element with buckets, when the elevator is stopped, is prevented by means of a holdback (noiseless ratchet or roller-type) mounted on the drive pulley (sprocket) shaft or in an elastic coupling between the electric motor and the input shaft of the reducer gear. In foreign designs, a hydraulic coupling is provided between the motor and reducer gear. In heavy-duty elevators, electromagnetic brakes are used as the holdback.

The casing head of elevators should be provided with tightly sealed manholes for inspection and repairs.

The take-up. Screw-type (Fig. 11.10), spring-screw and gravity take-ups are mostly employed in elevators. In a gravity take-up, the weight may act on the shaft of the take-up pulley directly (Fig. 11.11) or through a system of levers. A particular type of take-up is chosen depending on the type of the tractive element and drive, and the height of elevator. Elevators with round-link chains usually have a gravity-type take-up.

The take-up is arranged on the shaft of the lower pulley (or sprocket) and fastened to the side walls of the elevator boot. The take-up travel is 200-500 mm. The take-up pulley of belt elevators is made with a grid rim to prevent sticking of material particles. The diameter of a take-up pulley (or sprocket) is usually the same as that of the drive pulley.

The casing. The loading spout in the elevator boot may have a high or a low position (see Fig. 11.10). A high spout with the bottom inclined at 60° to the horizontal is employed for feeding of wet poorly flowing materials, and a low spout (with an angle of 45°),

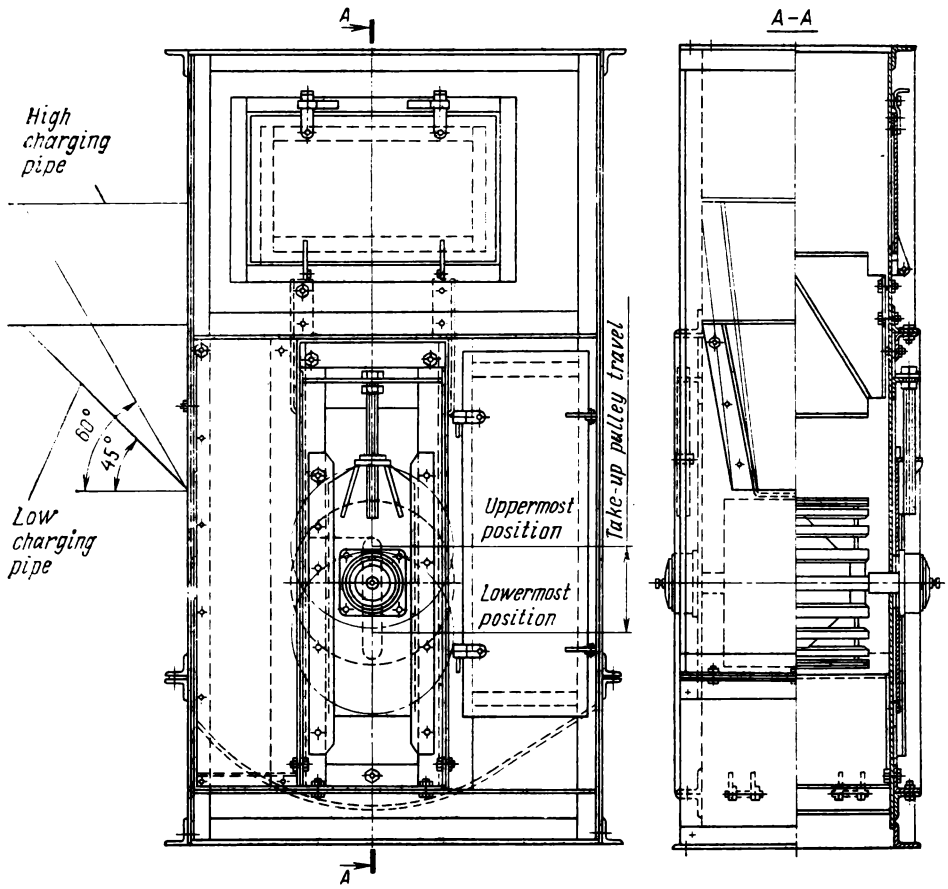


Fig. 11.10. Bottom part of casing (boot) and screw-type take-up of a vertical belt bucket elevator

for dry easily flowing loads. The boot has tightly closed manholes in the side walls for inspection and repairs. The mid sections of the elevator casing are made from steel sheets 2-4 mm thick and reinforced by steel angles longitudinally and around the connecting ends.

The height of sections is 2-2.5 m; they are connected together by bolts, with sealing washers laid in the joints. Elevator designs with a reinforced-concrete casing are known.

The casing is the supporting structure of the elevator; it absorbs static and dynamic forces. The moving part of an elevator is directed in the mid sections of the casing by guide means (see Fig. 11.1).

Safeguards. Falling of the moving part of an elevator on breakage of the chain or belt is prevented by special safeguards: chain catchers in chain elevators and steel cables in belt elevators. The cables are attached to the side walls of buckets and hang freely along the belt in normal operation, but take up the weight of the buckets if the belt breaks off occasionally. The take-up pulley (or sprocket) of an elevator may be additionally provided with a protective relay which de-energizes the drive motor on breakage of the tractive element.

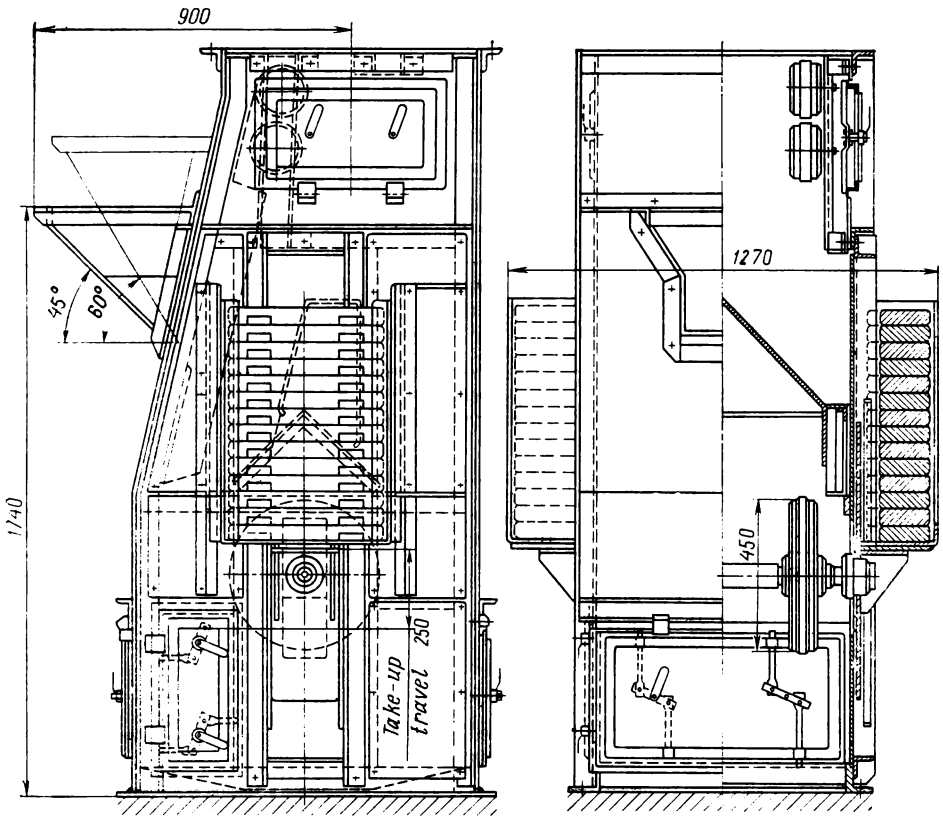


Fig. 11.11. Bottom part of casing (boot) and gravity take-up of a vertical chain bucket elevator

11.2.4. Elevator Calculation

Determination of the main characteristics. The type of elevator and the shape of buckets are chosen according to the characteristics of the materials to be transported (Table 11.3) and the required throughput capacity Q. The latter is found from the linear volume of buckets

$$i_0/t_b = Q/(3.6v\rho\psi) \qquad (11.10)$$

where i_0 is the useful geometrical volume of a bucket which may be taken from Table 11.1 or manufacturer's catalogue, l; t_b is the pitch of buckets, m [for widely spaced deep or shallow buckets, $t_b = (2.5\text{-}3.0)\,H$; for closely spaced buckets with guiding side walls, $t_b \approx H$, where H is the depth of bucket, m]; v is the speed of belt or chain, m/s (see Table 11.3); ρ is the density of material, t/m^3; and ψ is the coefficient of bucket filling (see Table 11.3).

For chain elevators, the bucket pitch should be a multiple of one or two chain pitches.

As follows from the relationship between the width and linear volume of standard buckets (Fig. 11.12), elevators with closely placed buckets can ensure a high throughput capacity even at reduced speeds, since they have a high i_0/t_b ratio.

For lumpy loads, the chosen buckets should be checked for the largest size of lumps, a

$$L \geqslant aX \qquad (11.11)$$

Table 11.3. Recommendations for Selecting Bucket Elevators

Kind of load	Typical examples	Type of elevator	Type of buckets	Average coefficient of bucket filling, ψ	Speed, m/s, of	
					belt	chain
Dry pulverized materials	Pulverized coal	Low-speed with gravity unloading	D	0.85	—	0.6-0.8
	Cement, rock phosphate meal	High-speed, centrifugal unloading	D	0.8	1.25-2.0	—
	Grain products (flour and formula feed)	High-speed, with centrifugal or gravity unloading	S	0.85	1-1.6	—
Pulverized and granular, wet poorly flowing	Soil, sand, powdered chalk, chemicals	High-speed, centrifugal unloading	S	0.6	1-2	0.8-2.0
Granular and fine-lump, low-abrasive	Food grain	High-speed, centrifugal unloading	D	0.75	2.0-3.2	—
	Sawdust, wood chips, dry clay in lumps, milled peat, fine coal	Ditto	D	0.8	1.25-2.0	1.0-1.6
	Lime, soot	Low-speed, gravity unloading	D	0.8	—	0.4-1.0
Granular and small-lump, highly abrasive	Gravel, ore, slags	Low-speed, directed gravity unloading	V R	0.8	0.4-0.8	0.4-0.63
	Sand, ash, soil, gangue	High-speed, centrifugal	D	0.8	1-2	—
Medium-size lumpy (a = = 60 mm), low-abrasive	Coal	Low-speed, directed gravity unloading	V R	0.7	—	0.4-0.63
	Dug peat	High-speed, centrifugal unloading	D	0.65	—	0.8-1.6
Medium-size lumpy, highly abrasive	Stone, ore, slags	Low-speed, directed gravity unloading	V R	0.7	—	0.4-0.63
Lumpy brittle materials	Charcoal	Low-speed, directed gravity unloading	V R	0.6	0.4-0.63	0.4-0.63

Note. Types of buckets: D–deep; S–shallow; V–V-shaped with side guides; R–round-bottom, with side guides.

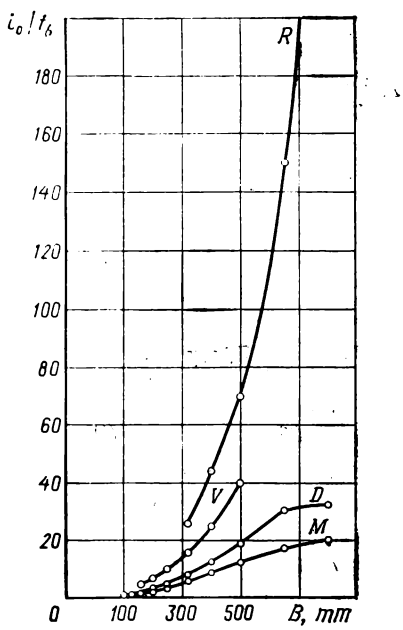


Fig. 11.12. Effect of bucket width B on i_0/t_b ratio
(bucket types: *M*—shallow; *D*—deep; *V*—V-shaped; *R*—round bottom)

where L is the length of bucket and X is a coefficient (X = 2-2.5 for ungraded loads and X = 4-5 for graded ones).

Pull calculation. The pull of an elevator is calculated by successive summation of the resistances in the elevator path. A specific feature of this calculation is that one has to consider the scooping resistance, i.e. the resistance offered by the material to buckets in the elevator boot. This resistance can be determined experimentally and depends on the characteristics of the transported material, the type, speed and spacing of buckets, the spacing between the buckets and walls of the boot, the coefficient of bucket filling, and the method of bucket filling.

The scooping resistance can be roughly determined by the formula:

$$W_{sc} = K_{sc} q_l \qquad (11.12)$$

where K_{sc} is the scooping coefficient, or the unit work spent for scooping, Nm/N; and $q_l = gQ/(3.6v)$ is the linear gravity force of the load, N/m.

With the bucket speed of 1-1.25 m/s, it may be recommended to take K_{sc} = = 1.25-2.5 N m/N for pulverized and fine-size lumpy loads and K_{sc} = 2-4 N m/N for those with medium-size lumps. The masses of the belt, chains and buckets are taken from manufacturer's certificates or corresponding standards.

For friction-drive elevators, the highest tension $S_{max} = S_{on}$ should be checked by engagement of the belt with the pulley using the general theory of frictional drive.

The highest rated tension thus found is used to determine the strength of the chain or belt (see Chs. 3 and 4). The power of the drive motor is found by formula (2.45).

Example. Calculate a vertical bucket elevator (Fig. 11.13*a*) for transportation of used moulding sand in a foundry shop.

Initial data. The rated throughput capacity Q = 30 t/h, lifting height H = 22 m, and sand density ρ = = 1.5 t/m^3.

Calculation. The main characteristics of the elevator. We find from Table 11.3 that a belt elevator with widely spaced deep buckets and belt speed v = 1.6 m/s is suitable for the purpose. We select a four-ply belt; then the diameter of the drive pulley D_p = 125 × 4 = 500 mm, or its radius r_p = 0.25 m. The rotational speed of the pulley at v = = 1.6 m/s.

$$n = 60v/(\pi D_p) = 60 \times 1.6/(3.14 \times \times 0.5) = 61 \text{ rpm}$$

The pole distance is found by formula (11.3)

$$h_p = 895/n^2 = 895/61^2 = 0.24 \text{ m}$$

Since $h_p < r_p$, the buckets can be unloaded centrifugally. By formula (11.3) at ψ = 0.8 we have

$$i_0/t_b = Q/(3.6v\rho\psi) = 30/(3.6 \times \times 1.6 \times 1.5 \times 0.8) = 4.34 \text{ l/m}$$

Referring to Table 11.2, we choose (for $i_0/t_b = 5$) deep-type buckets with a width $B = 250$ mm and volume 2 l; then the belt width $B_b = 300$ mm and bucket pitch $t_b = 0.4$ m.

With these characteristics of the buckets and the belt speed 1.6 m/s, the rated throughput capacity of 30 t/h can be obtained at the coefficient of bucket filling

$$\psi = t_b Q/(3.6v\rho i_0) = 0.4 \times \times 30/(3.6 \times 1.6 \times 1.5 \times 2) = 0.7$$

It would be possible to reduce the speed to 1.4 m/s, but it is unstandard and cannot ensure centrifugal discharge.

Linear gravity forces. The linear mass of the belt (four-ply belt type 2 of fabric БКНЛ-65) is $m_{rb} = 2.55$ kg/m. Then

$$q_{rb} = gm_{rb} \approx 9.81 \times 2.55 \approx 25 \text{ N/m}$$

The mass of a standard deep bucket ($B = 250$ mm) is $m_b = 3.38$ kg, then

$$q_0 = q_{rb} + gm_b/t_b = 25 + 9.81 \times 3.38/0.4 = 108 \text{ N/m}$$

The useful load

$$q_l = gQ/(3.6v) = 9.81 \times \times 30/(3.6 \times 1.6) = 52 \text{ N/m}$$

The linear gravity force acting on the carrying runs

$$q = q_0 + q_l = 108 + 52 = 160 \text{ N/m}$$

Pull calculation. According to the diagram of tensions (Fig. 11.13*b*), the lowest tension of the belt should be expected at point *1*. We can make only a rough calculation, since we do not know the tension S_4 in the belt section running off the drive pulley as required to ensure the rated pull. We assume that $S_1 = S_0$. The tension at point *2* is found by considering the resistance on the take-up pulley and the scooping resistance by formula

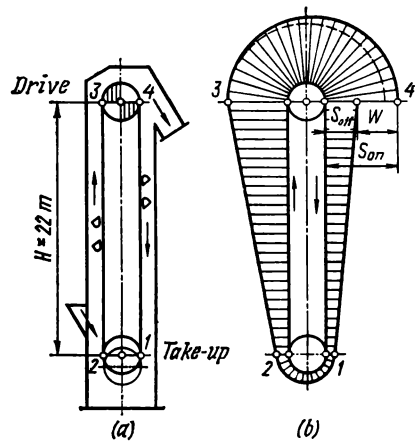


Fig. 11.13. Diagram for elevator calculation (*a*) elevator scheme; (*b*) diagram of belt tensions

(11.15) for heavy-duty operation at $\xi = 1.08$ and $K_{sc} = 2$

$$S_2 = \xi S_1 + W_{sc} = 1.08S_0 + K_{sc}q_l = 1.08S_0 + 2 \times 52 = 1.08S_0 + 104$$

$$S_3 = S_{on} = S_2 + qH = 1.08S_0 + 104 + 160 \times 22 = 1.08S_0 + 3630$$

Calculation against the direction of belt motion gives

$$S_4 = S_{off} = S_1 + q_0 H = S_0 + 108 \times 22 = S_0 + 2380$$

According to the theory of frictional drives, we have

$$S_{on} \leqslant S_{off} e^{\mu\alpha}$$

or for the case considered

$$S_3 \leqslant S_4 e^{\mu\alpha}$$

For a steel pulley and high humidity (the elevator operates outdoors), the friction coefficient is $\mu = 0.1$, so that at $\alpha = 180°$ we have $e^{\mu\alpha} = 1.37$. Hence,

$$S_3 \leqslant 1.37S_4$$

or for the case considered

$$1.08S_0 + 3630 \leqslant 1.37\,(S_0 + 2380)$$

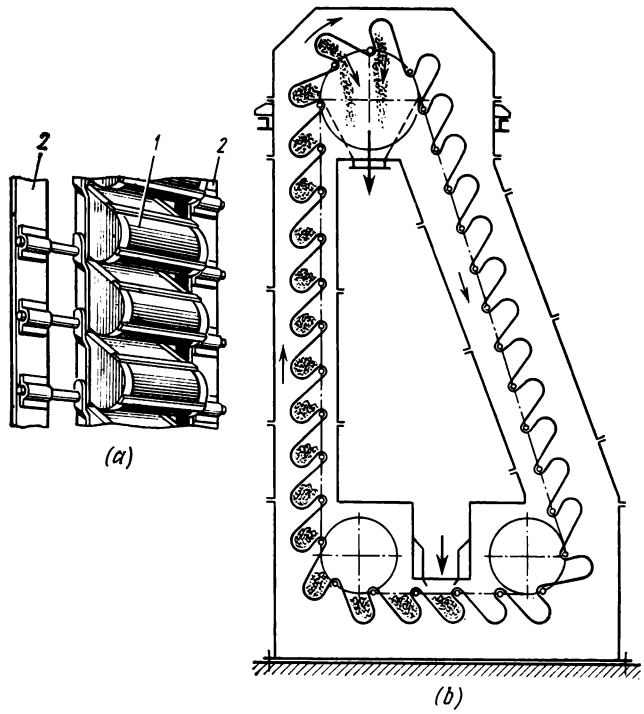


Fig. 11.14. Schenck elevator (FRG)
(a) buckets; (b) elevator path

Solution of this equation gives $S_0 > 290$ N. To ensure a certain margin, we take $S_0 = 500$ N and then

$$S_3 = S_{on} = 1.37S_4 = 1.37 \times \times 2880 \approx 4000 \text{ N}$$

$$S_4 = S_{off} = S_0 + 2380 = 2880 \text{ N}$$

The required number of belt plies is found (see Ch. 4) for $K_{op} = 55$ N/mm and safety factor $K = 10$

$$i = KS_{max}/BK_{op} = 10 \times \times 4000/(300 \times 55) = 2.43$$

Noting that the belt is weakened by bolts and should ensure firm fastening of buckets, we can use the four-ply belt adopted earlier in the calculation.

The circumferential force on the drive pulley, noting the pulley resistance, is

$$W = (S_3 - S_4)\,\xi = (4000 - 2880) \times 1.08 = 1210 \text{ N}$$

The power of the drive motor, assuming the efficiency of the drive mechanism $\eta = 0.85$ and a margin $K = 1.25$, will be

$$N = KWv/(1000\eta) = 1.25 \times \times 1210 \times 1.6/(1000 \times 0.85) = 2.8 \text{ kW}$$

We take finally a drive motor of a power of 3 kW and rotational speed 980 rpm.

11.3. Special Types of Bucket Elevators

Special-type bucket elevators for transportation of abrasive (ore), brittle (pellets, briquettes) and hot bulk materials (clinker and sinter with temperatures up to 900°C) have been developed by Schenck and Beumer Cor. (the FRG). They have 'pocket-shaped' buckets and central gravity loading and unloading (Fig. 11.14). Buckets *1* (Fig. 11.14*a*) are made of rubber, plastics or steel (for hot loads) and faste-

ned close to one another on two narrow belts *2* or round-link chains.

An advantage of this type of elevators is that the material is not scooped by the buckets, but fed into them from the top, which decreases the resistance in loading and prevents the material from crushing. A drawback is a rather intricate design, especially of elevators with two parallel belts.

The main characteristics of these elevators are as follows: bucket width 200-1600 mm, bucket length 400-710 mm, bucket pitch (closely placed buckets) 250-710 mm, lifting height up to 55 m, and throughput capacity 5-1000 m^3/h at a speed of 0.3-0.4 m/s.

11.4. Shelf Elevators

Shelf elevators may be of vertical (Fig. 11.15*a* and *c*) or inclined type (Fig. 11.15*b*). They are used for transportation of various unit loads (barrels, boxes, bales, machine parts, etc.).

A shelf elevator (Fig. 11.15*a*) has two vertical endless chains running around a top and a bottom sprocket. The chains carry cantilever shelves (or racks) which are shaped according to the kind of loads and fastened to the chains with a definite spacing from one another (Fig. 11.16).

The shelves are loaded and unloaded automatically or manually. Automatic loading and unloading is performed most simply with cylindrical load units which can be rolled on an inclined table and easily reloaded from a rack table on shelves and vice versa. Unloading is usually performed at the top point of the descending run, though it is also possible to unload the shelves in the ascending run by tilting them on snub sprockets, (see Fig. 11.15*c*).

The load-carrying racks are made in the form of straight or curved brackets (Fig. 11.16*a*), according to the shape of loads. In some cases, the racks are attached pivotedly and can be unloaded at any point of the ascending run by means of stops or guide bars. After being unloaded, the racks are returned to the initial position by means of springs or guide bars.

The tractive element in shelf elevators is made in the form of laminated sleeve-type or wheel chains. Shelf elevators have low speeds (not more than 0.2-0.3 m/s).

The pull of a shelf elevator is calculated in the conventional way (see Ch. 2). One only has to consider an additional resistance of chain wheels on stationary guides, which appears owing to the cantilever position of the load relative to the chains (Fig. 11.15*d*). If m is the mass of a load unit on a shelf, l is the distance from the common centre of gravity of the shelf and load to the upper point of shelf attachment, and b is the vertical distance between the upper and lower points of shelf attachment, the pressure on the guides (neglecting the effect of chain tension) will be

$$N_p = gml/b \qquad (11.13)$$

and the additional resistance

$$W_{ad} \approx 2N_p w = 2gmlw/b \qquad (11.14)$$

where w is the coefficient of resistance of chain wheels in guides.

The total force required for vertical lifting of a loaded shelf

$$P_v = gm + W_{ad} = gm\,(1 + 2lw/b) \qquad (11.15)$$

If the hauling chains of an elevator are provided with support wheels, $w = 0.03\text{-}0.05$; for chains without wheels which slide in guides, $w = f = 0.25\text{-}0.35$, where f is the coefficient of friction of the chain in guides; it depends on the operating conditions of an elevator.

11.5. Cradle Elevators

Cradle elevators (Fig. 11.17) are employed for vertical lifting of various

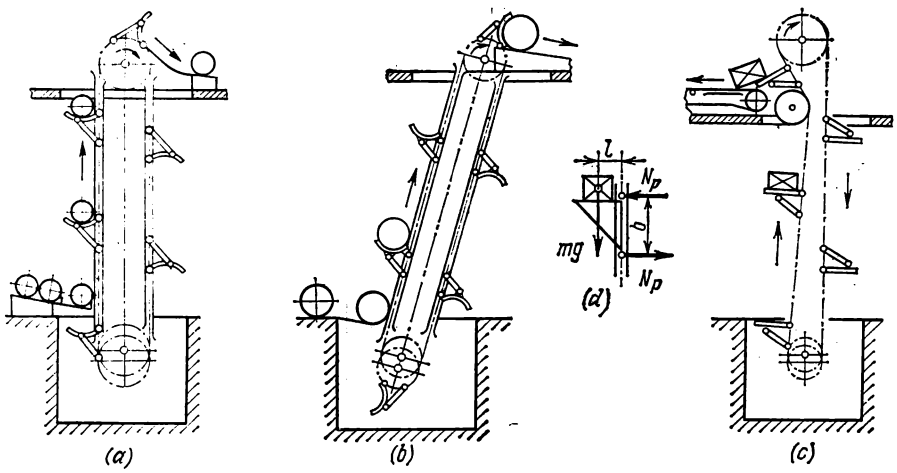


Fig. 11.15. Shelf elevators
(*a*) vertical; (*b*) inclined; (*c*) vertical with clamp deflection in the carrying run; (*d*) diagram of forces

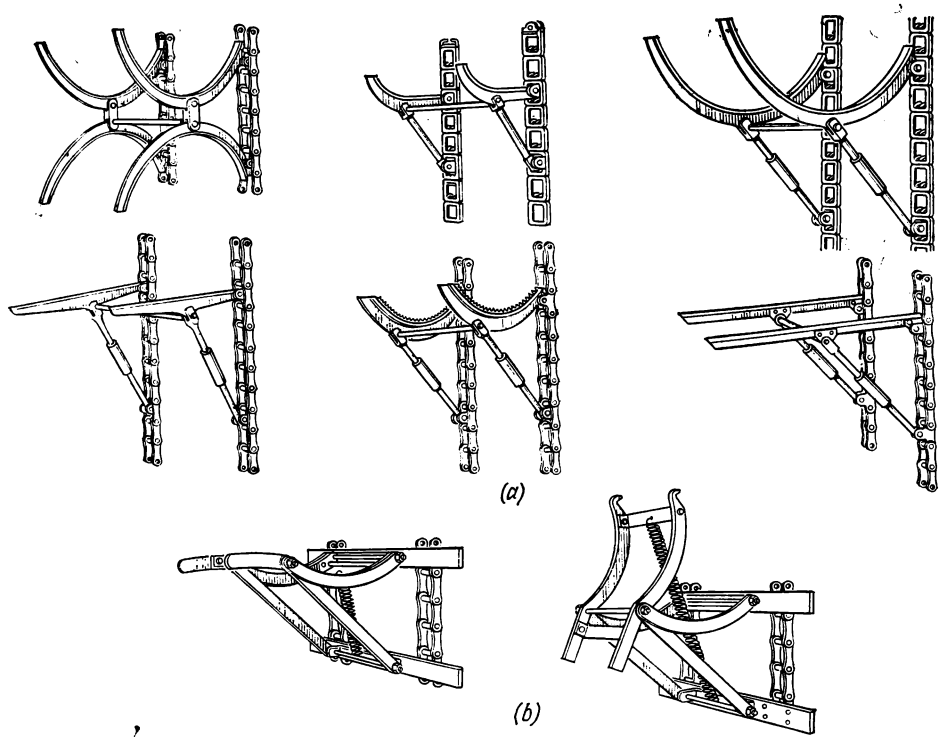


Fig. 11.16. Racks of various shelf elevators

unit loads. In contrast to shelf elevators, their load-carrying elements (called 'cradles') are pivotedly suspended from the chains, which makes it possible to unload them at any point of the descending run of the elevator.

Cradle elevators have essentially the same elements as cradle conveyers (see

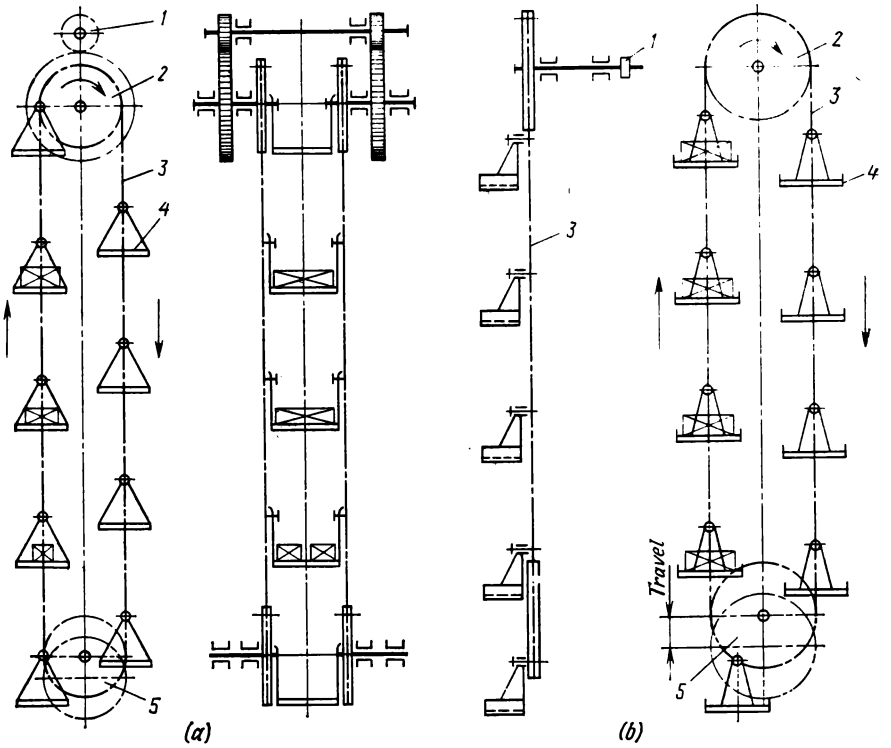


Fig. 11.17. Schemes of cradle elevators
(*a*) double-chain; (*b*) single-chain; *1*—drive; *2*—drive sprockets; *3*—hauling chains; *4*—cradles; *5*—take-up sprockets

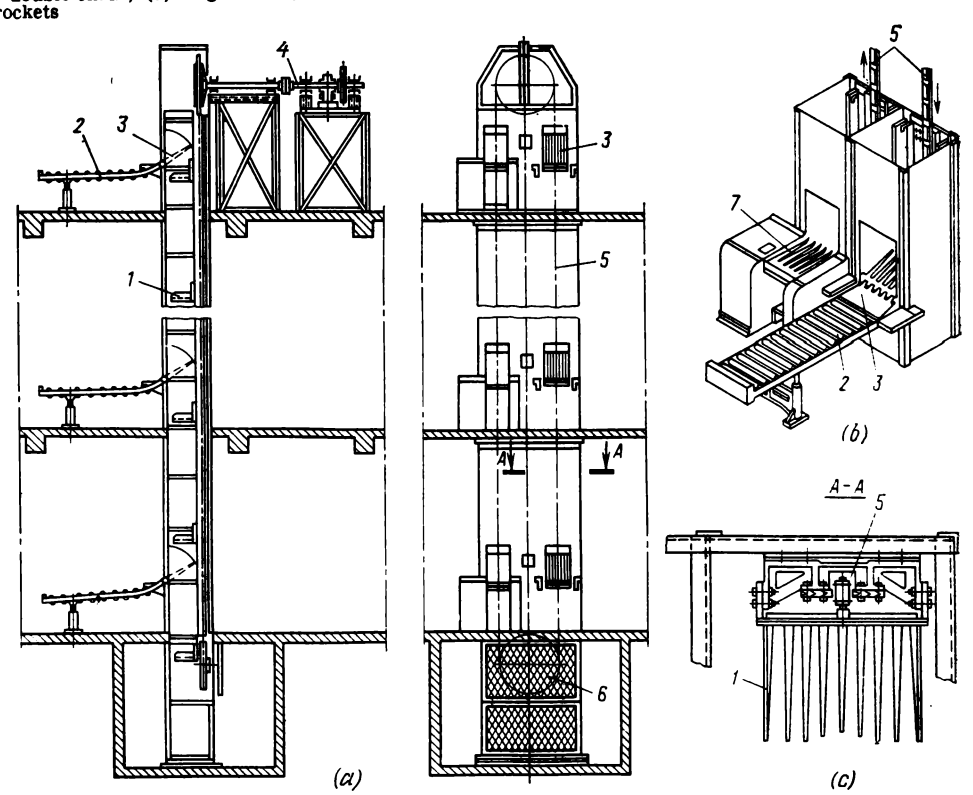


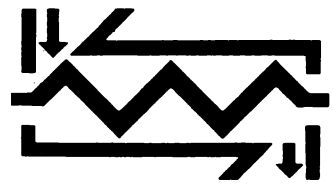
Fig. 11.18. Single-chain cradle elevator for books
(*a*) general view; (*b*) loading and unloading devices; (*c*) cantilever cradle; *1*—cradle; *2*—roller conveyer; *3*—unloading table; *4*—drive; *5*—hauling chain; *6*—take-up; *7*—loading grate table

Ch. 7) and differ from these only in the path configuration; cradle elevators carry loads only vertically, while conveyers transport the loads along a path consisting of horizontal and vertical sections. Cradle elevators are usually of the double-chain type (Fig. 11.17*a*). Single-chain elevators with cantilever cradles (Fig. 11.17*b*) are employed for transportation of light loads, such as books. A cradle elevator (Fig. 11.18) is provided with automatic loading and unloading means on each floor of the building. On pressing a control button, the grate table with a load moves into the elevator shaft and a rack-shaped cradle takes up the load from it. After that, the table is automatically retracted from the shaft to the initial position. At an unloading station, the turn-table turns by a control signal into the shaft and a descending cradle leaves its load on it. The load slides on an inclined table onto a roller conveyer and is delivered to the destination point, after which the turn-table returns to the initial position.

Cradle elevators are calculated in the same way as cradle conveyers (see Chs. 2 and 7).

12

Screw Conveyers



12.1. Types and Applications

Screw conveyers exist in three main types: horizontal and slightly inclined at an angle up to 20° (the main type), vertical and steeply inclined, and conveying screw pipes. The three types have certain features in common, but differ in other features and especially in the operating principle.

Screw conveyers are employed for transportation of pulverized, granular and, less frequently, small-size lumpy bulk materials to relatively short distances (usually up to 40 m horizontally or to heights up to 30 m) in the chemical, flour industries, and in the manufacture of building materials.

Screw conveyers are poorly applicable with sticky, slumping and abrasive materials.

Among the advantages of screw conveyers are simple design, easy maintenance, small dimensions, convenient intermediate unloading, and tightness which is of especially high value when conveying dusty, hot or odorous loads. Their drawbacks are a high unit power consumption, substantial rubbing and crushing of the conveyed material, high wear of the screw and trough, and a high sensitivity to overloads which can lead to load accumulation in the trough (especially at the bearings).

12.2. Horizontal Screw Conveyers

A horizontal (or slightly inclined) screw conveyer (Fig. 12.1) has a screw arranged on a bearing-supported shaft, a trough with a semicylindrical bottom arranged coaxially with the screw, and a drive (electric motor and reducer gear) to rotate the shaft with screw. A bulk material is fed to the trough through one or more openings in its cover; as the screw rotates, the material is propelled along the shaft, as a nut would be moved on a threaded bar if retained from rotation together with the bar. Rotation of the material together with the screw is prevented by its gravity force and friction on the trough walls. The material is unloaded from the conveyer through one or more openings in the bottom, which are provided with gates.

The conveyer screw may be right- or left-handed, with one, two or three leads. The screw surface may be continuous or discontinuous: band-like or in the form of separate blades (Fig. 12.2). Continuous screws are used mainly for conveying dry fine-granular and pulverized materials not liable to slumping; discontinous screws are effective with slumping materials. Besides, blade- and band-like screws are used in cases when

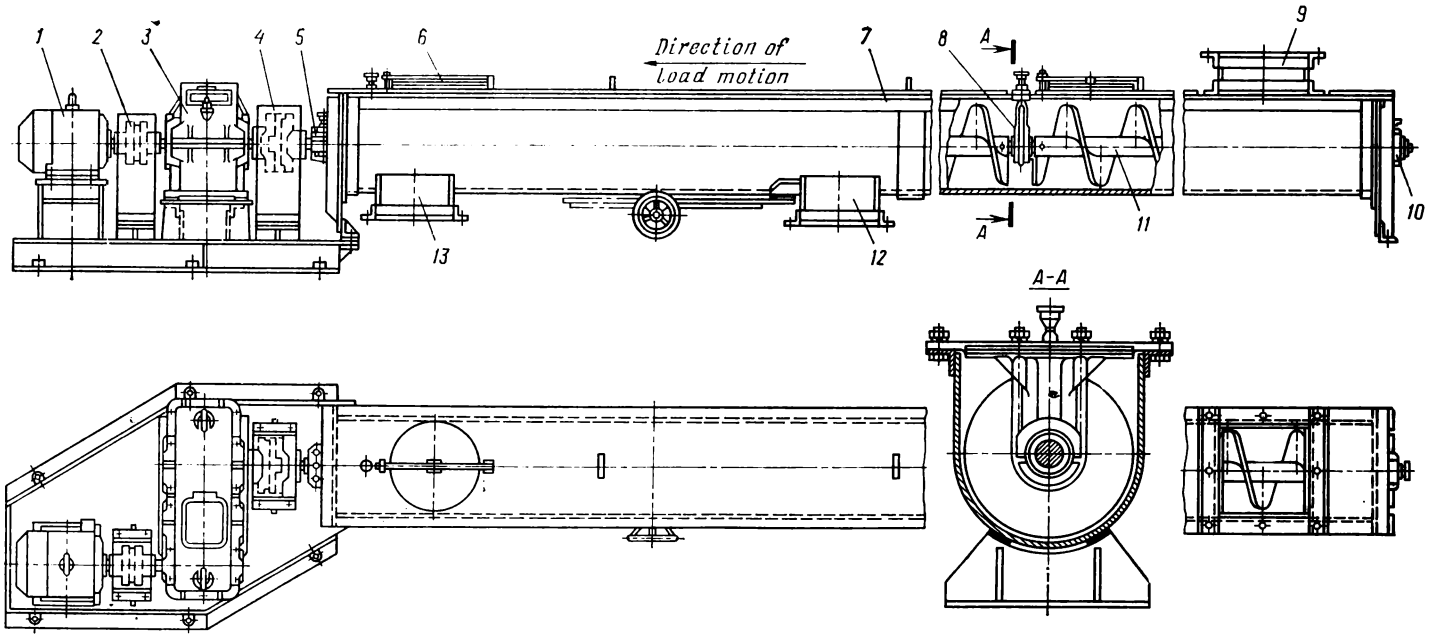


Fig. 12.1. Screw conveyer

1—electric motor; *2*, *4*—couplings; *3*—reducer gear; *5* — head bearing; *6*—viewhole; *7*—trough; *8*—intermediate bearing; *9*—charging pipe connection; *10*—rear bearing; *11*—screw shaft; *12*—intermediate unloading pipe; *13*—front unloading pipe

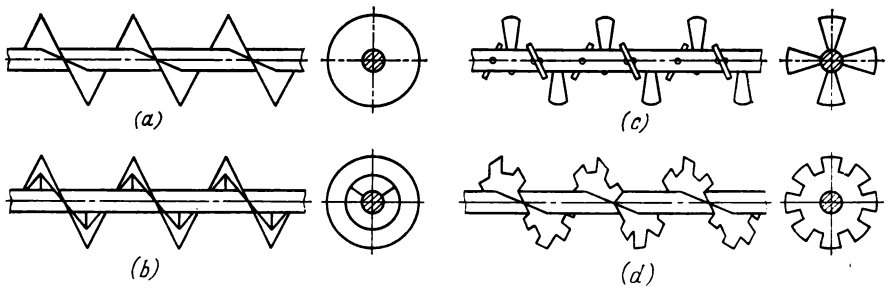


Fig. 12.2. Screws of screw conveyers
(*a*) continuous; (*b*) band-like; (*c*) bladed; (*d*) special-shape

thorough intermixing of the material during conveying is essential.

The helix of continuous and band-like screws is made by stamping from steel sheets or strips, which is then welded to the shaft. The helix of a band-like screw and blades are fastened by pins entering drilled holes in the shaft.

The screw shaft may be solid or tubular and consist of a number of sections for ease of assembly. Tubular shafts have a lower mass and their sections can be connected together more conveniently by means of short shafts inserted into their ends. The screw shaft is supported by end and intermediate bearings. Intermediate bearings are suspended from cross-plates fastened to the trough. Their diameter and length should be as small as possible (since the helix is interrupted on them). The bearings should be sealed properly against fouling by material particles. In some cases, the shaft is supported in the mid by the connecting shafts which rotate in sliding bearings. The lubricant is fed to the bearings through pipes from grease cups mounted on plates above

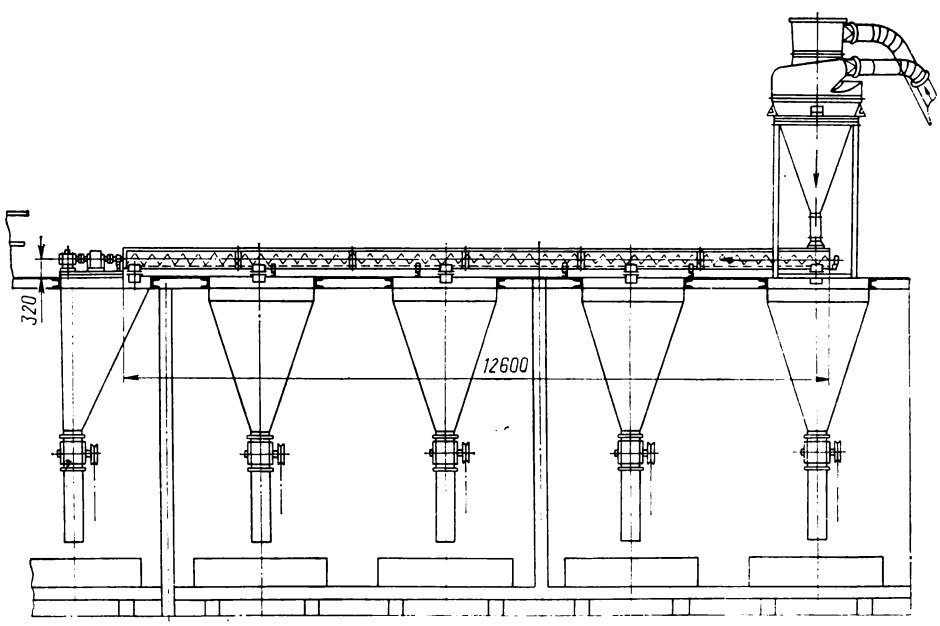


Fig. 12.3. Arrangement of screw conveyer above sand-preparing bunkers of a foundry shop

them. The end bearings are mounted in the end walls of the trough. One of them is of the thrust type (at the conveyer end) to absorb the axial tensile force acting along the shaft.

Figure 12.3 illustrates the application of a screw conveyer for distribution of dried foundry sand between bunkers. The material is supplied from the drying section of a foundry shop by pneumatic transport into a cyclone from which the sand is poured through a feeding gate into the conveyer trough.

The throughput capacity of a screw conveyer (Q, t/h, or V, m³/h) depends on the screw diameter D (m), lead of screw t (m), rotational speed n (rpm) and the coefficient of filling of screw cross section ψ

$$Q = V\rho = 60\frac{\pi D^2}{4} tn\psi\rho C = 47D^2 tn\psi\rho C \qquad (12.1)$$

where ρ is the density of the material, t/m³, and C is a correction factor depending on the angle of incline β of the conveyer

β, deg .	0	5	10	15	20
C . . .	1	0.9	0.8	0.7	0.6

The conveying speed, m/s

$$v = tn/60 \qquad (12.2)$$

The filling coefficient ψ is taken at a relatively low value to avoid material accumulation on intermediate bearings; it depends on the properties of conveyed materials (Table 12.1).

Table 12.1. Values of Coefficients ψ, A and w for Screw Conveyers

Kind of load	Coefficient		
	ψ	A	w
Light non-abrasive (grain products, flour sawdust)	0.4	65	1.2
Light slightly abrasive (chalk, pulverized coal, asbestos, peat, soda)	0.32	50	1.6
Heavy low-abrasive (salt, lumpy coal, dry clay)	0.25	45	2.5
Heavy abrasive (cement, ash, sand, wet clay, crushed ore, slag)	0.125	30	4.0

In inclined screw conveyers with ascending motion of the material, the throughput capacity decreases with increasing angle of incline, especially if the shaft is supported by intermediate bearings.

For easily movable materials, the lead of screw is usually taken as $t = D$; for poorly moving materials, a smaller screw lead is taken (roughly $t = 0.8D$) or a screw with two or three leads is employed.

The rotational speed n of a screw is chosen, depending on the kind of conveyed material and screw diameter; it should ensure smooth flow of the material without overflowing over the shaft; the rotational speed is diminished with an increase of screw diameter and the density and abrasiveness of the load. The highest allowable rotational speed (rpm) of a screw can be found by the empirical formula

$$n_{max} = A/\sqrt{D} \qquad (12.3)$$

where A is a coefficient (see Table 12.1), and D is the screw diameter, m.

The value of D in equation (12.3) is taken approximately and then checked by equation (12.1). It is finally chosen from the standard series of screw diameters: 0.1, 0.125, 0.16, 0.2, 0.25, 0.32, 0.4, 0.5, 0.63, and 0.8 m.

The diameter of screws for conveying of lumpy loads should be selected depending on the size of lumps

$$D \geqslant (10\text{-}12)\,a; \quad D \geqslant (4\text{-}6)\,a_{max} \qquad (12.4)$$

where a is the size of lumps of graded (of uniform lump size) material and a_{max} is the size of largest lumps in an ungraded material.

The total resistance to the motion of material in a screw conveyer is made up of friction of the load on the trough and screw surface, resistances in intermediate and end bearings (including

the thrust bearing), and resistances to the ascending motion. These resistances can be calculated readily, if the friction coefficients are known. There are, however, some other additional resistances which can hardly be calculated. They include resistances due to accumulation of the material on intermediate bearings, friction of the screw edges on the material drawn into the gap between them and the trough, and intermixing of the material. For that reason, the power of the conveyer motor is determined more simply by using an experimental total coefficient of resistance w which mainly depends on the properties of the conveyed material (see Table 12.1). The power on the motor shaft N, kW, is found by substituting an appropriate value of w into formula (2.18).

The torque on the screw shaft (N m) is determined by the formula

$$M_0 = 1000 \times 60N\eta/(2\pi n) \qquad (12.5)$$

where N is in kilowatts, η is the efficiency of the drive mechanism, and n is the rotational speed, rpm.

The highest longitudinal force (N) acting on the screw

$$P = \frac{M_0}{r \tan(\alpha+\varphi)} = \frac{2M_0}{kD \tan(\alpha+\varphi)} \qquad (12.6)$$

where r is the radius of application of force P, m; it can be roughly taken that $r = kD/2 = (0.7\text{-}0.8)\, D/2$; α is the angle of rise of the helix line at radius r; φ is the angle of friction of the material on the screw surface; and $\tan\varphi = f_1$ (where f_1 is the coefficient of friction of the material on the screw surface).

The screw shaft, which can be regarded as a divided multiple bearing shaft with a distance l between the supports (bearings), is calculated for a complex resistance to torsion by the torque M_0, longitudinal compression or tension by force P, bending under the moment Pr, and bending along the length l under the gravity force. The screw is also checked for bending deformation which should not change substantially the gap between the screw edges and the trough.

12.3. Vertical Screw Conveyers

Vertical screw conveyers relate to special types (Fig. 12.4*a*, and *b*). The conveyer has a shaft with a continuous helix, which is suspended on a thrust bearing and rotates in a cylindrical casing (tube), and a short horizontal screw feeder which rotates in a short pipe connection. The two screws may be driven from a common drive (Fig. 12.4*a*) or separately (Fig. 12.4*b*). The conveyer is unloaded through a pipe connection at the top of the casing. The bottom portion of the vertical screw, which receives the fed material, is made with a varying (decreasing upward) diameter or with a smaller screw lead. With a large height of conveyer, intermediate bearings are sometimes provided to prevent radial run-out of the shaft.

A steep-slope overloading screw conveyer with a flange-mounted electric motor is illustrated in Fig. 12.4*c*.

The upward motion of bulk material along the vertical screw occurs as follows. As the material is fed by a horizontal feeder screw to the vertical screw, it is rotated by the latter. The material is pressed by centrifugal forces to the cylindrical walls of the conveyer casing. Due to the forces of gravity and friction, it lags somewhat behind the motion of the helix, i.e. rotates at a smaller angular speed than that of the screw; therefore, it is moved upwards along the screw axis, similar to a nut moving on a thread, along a helical path, but with a lower axial speed than the speed of a retained nut which would move along the screw.

In order to develop a sufficiently high centrifugal force, the screw should rotate with a sufficient frequency, i.e. in contrast to horizontal screws whose rotational speed is limited by the highest value, the speed of vertical screws

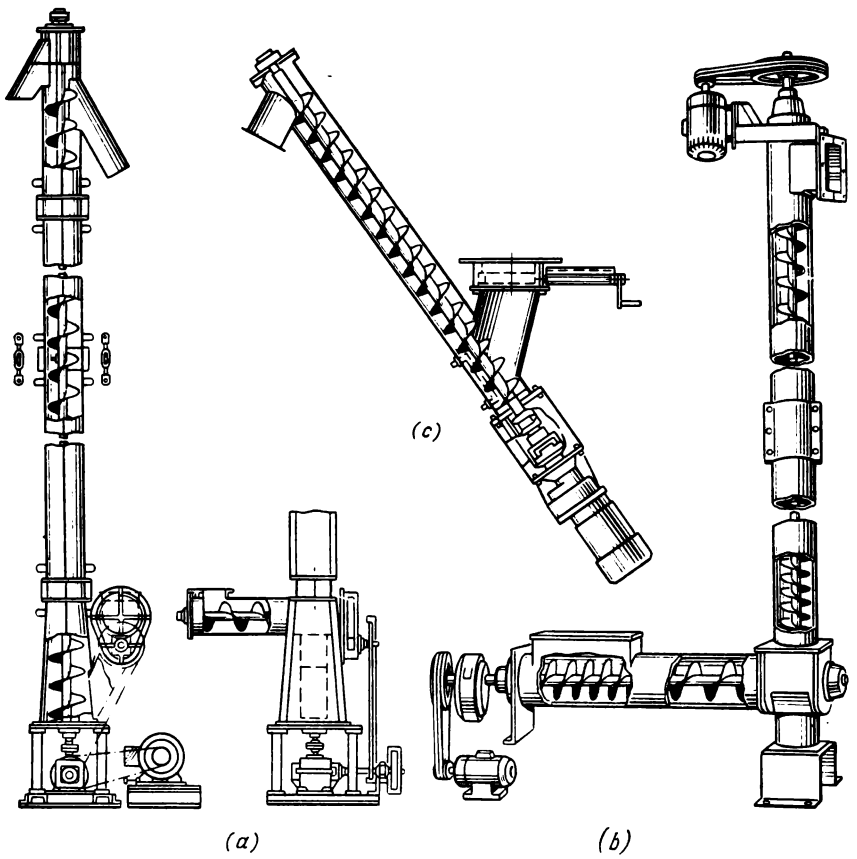


Fig. 12.4. Screw conveyers
a), (b) vertical; (c) steep-slope

should be not less than an allowable (critical) lowest value which is required to move the material upwards along the screw. Let us see how this speed can be determined. The gravity force G acting on a particle on the helix surface can be resolved into two components: a force $G \sin \alpha$ directed along the helix surface downward and a force $Gf_h \cos \alpha$, the force of friction of the particle on the helix surface (f_h is the coefficient of friction), see Fig. 12.5.

When a particle rotates at the screw circumference with a velocity v, there appears a centrifugal force mv^2/R (where m is the mass of particle), and therefore, the force of friction of the particle on the cylindrical wall of the casing,

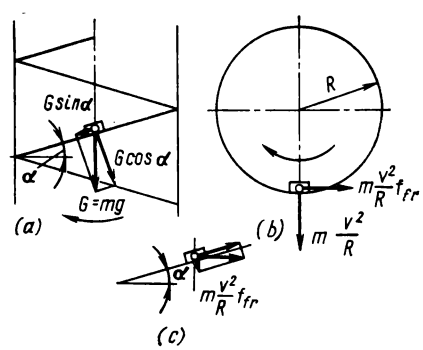


Fig. 12.5. Diagram for calculation of a vertical screw conveyer

which acts in the horizontal plane, is equal to $mf_c v^2/R$ where f_c is the coefficient of friction of the particle on

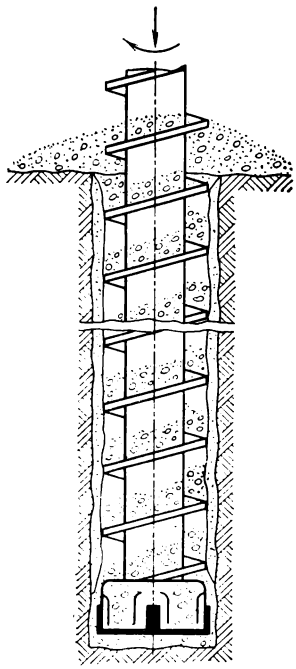


Fig. 12.6. Drilling rod with conveying screw

the casing (Fig. 12.5*b*). The component of this force which acts on the helix surface is $mv^2 f_c \sin \alpha/R$ (Fig. 12.5*c*) and the force of friction on the helix surface i $mv^2 f_c f_h \sin \alpha/R$.

The highest velocity at the screw circumference at which a particle has no axial motion, but rotates together with the screw, i.e. the critical speed v_{cr}, can be found from the equation

$$mg \sin \alpha + mg \cos \alpha f_h + mv_{cr}^2 f_c f_h \sin \alpha/R - mv_{cr}^2 f_c \cos \alpha/R = 0 \quad (12.7)$$

whence

$$v_{cr} = \sqrt{\frac{gR \sin \alpha + f_h \cos \alpha}{f_c \cos \alpha - f_h \sin \alpha}} = \sqrt{\frac{gR}{f_c} \tan (\alpha + \rho_h)} \quad (12.8)$$

where $\rho_h = \arctan f_h$.

Noting that $v_{cr} = 2\pi Rn_{cr} = \pi Dn_{cr}$, the critical speed of screw rotation

$$n_{cr} = \frac{30}{\pi} \sqrt{\frac{2g}{Df_c} \tan (\alpha + \rho_h)} \quad (12.9)$$

At rotational speeds $n > n_{cr}$, particles of the material will move upwards along the helix surface.

As may be seen from formula (12.9), for a material particle to move upwards along the helix axis, its rotational speed should be higher at a higher helix angle of the screw, smaller screw radius, lower coefficient of friction of the particle on the screw, and higher coefficient of friction on the casing wall.

Vertical screw conveyers for transportation of powdered, granular and fine-size lumpy bulk materials have a limited throughput capacity and a lifting height up to 15 m or rarely, up to 30 m. Their advantages and drawbacks are essentially the same as those of horizontal screw conveyers. As compared with bucket elevators, they have smaller dimensions and allow more convenient unloading in any direction, but require an appreciably higher power of the drive.

A special case of application of vertical screw conveyers (with a rather small screw diameter and low throughput capacity) is drilling of boreholes and pits. In that case, the load-carrying screw element is combined with the drilling member, a quickly rotating drill rod with drill bit at the end (Fig. 12.6). The role of conveyer casing is then played by the cylindrical walls of

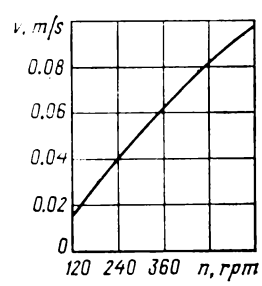


Fig. 12.7. Effect of rotational speed of drill rod on conveying speed

the borehole or a casing pipe. Fine loose particles accumulated under the drill bitin drilling are caught up and carried upwards by a screw mounted on the drill rod. The speed of conveying and the throughput capacity increase with an increase of the rotational speed of the rod (Fig. 12.7).

12.4. Conveying Screw Pipes

A conveying screw pipe (Fig. 12.8) rotates on two pairs of support rollers arranged a certain distance apart. A helical is attached to the inside surface of the pipe. As the pipe rotates, a bulk material fed into it at one end is poured over the helix edges and propagates gradually along the pipe axis and is discharged at the other end. Two steel rings are set on the pipe to contact the support rollers. In addition to the support rollers, the pipe supports have two pairs of thrust rollers arranged at both sides of a ring; this prevents the pipe from axial displacement. The pipe is rotated from a gear train, the last gear in the train being set on the pipe. A short conical-screw section may be provided at the charging end for convenient feeding of the material into a small-diameter pipe (Fig. 12.8*a*). The material is discharged by pouring over the pipe edge at the discharging end.

Conveying screw pipes are arranged horizontally or with a slight incline. A version of this type of conveyer is a conveying pipe which has no screw inside, but a smooth cylindrical surface. To effect motion of material, the pipe is always mounted with a slight decline towards the discharge end. Pipes of this type are mostly used for performing various process operations, such as calcining, drying, sintering, intermixing, etc. The length of a pipe, which sometimes may be as large as 200-250 m, is often determined by the time required for a process operation.

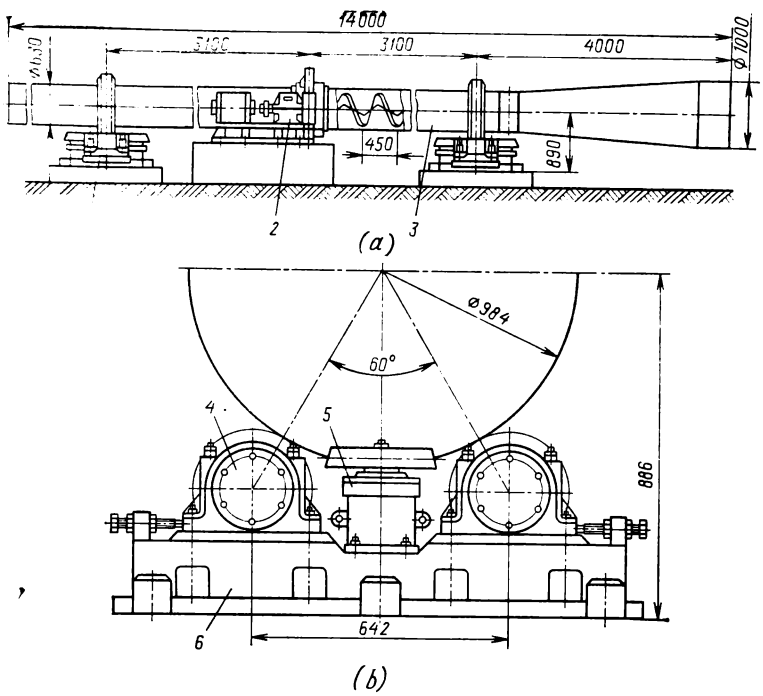


Fig. 12.8. Conveying screw pipe
(*a*) general view; (*b*) support frame with rollers; *1*—radial-thrust rollers; *2*—drive; *3*—pipe; *4*—support roller; *5*—thrust roller; *6*—frame

The advantages of conveying screw pipes include a rather simple design, high reliability, isolation from the surroundings along the entire pipe length, and wide range of throughput capacities and lengths. Among their drawbacks are a high mass, substantial overall dimensions, and high power consumption (which is however smaller than in screw conveyers).

The throughput capacity of a conveying screw pipe can be determined by formula (12.1) and the conveying speed v, by formula (12.2). If the load should remain in the pipe for processing during a time t, the required length of the pipe is

$$L = vt \qquad (12.10)$$

The screw lead is usually taken equal to $0.5D$ and the filling coefficient $\psi = 0.2\text{-}0.3$.

The material will move in a pipe, if the speed of pipe rotation does not exceed a certain critical value n_{cr}; otherwise, the material will be held at the internal surface of the pipe by centrifugal force C and will rotate together with the pipe. In order to determine n_{cr}, let us consider the condition of equilibrium for a particle which is acted upon by the gravity force G and has been raised in the rotating pipe by an angle α.

The particle in the pipe will begin to slide on the pipe surface when the force of its friction on the pipe wall becomes smaller than the component of the gravity force of the particle, directed along the tangent to the surface (Fig. 12.9*a*). Upon substitution of $\omega = \pi n/30$ into the equation

$$G \sin \alpha = (C + G \cos \alpha) f$$

$$= \left(\frac{GD}{2g} \omega^2 + G \cos \alpha \right) f \qquad (12.11)$$

(where $f = \tan \varphi$ is the coefficient of friction of the particle on the pipe surface), we obtain

$$n = 42.3 \sqrt{\frac{\sin (\alpha - \varphi)}{D \sin \varphi}} \qquad (12.12)$$

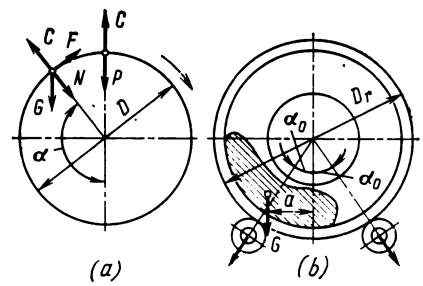


Fig. 12.9. Diagram for calculation of a conveying screw pipe

The critical speed of rotation $n_{cr} = n_{max}$ is found by substituting $\alpha = 90° + \varphi$

$$n_{cr} = 42.3 \sqrt{1/(D \sin \varphi)} \qquad (12.13)$$

At the top point ($\alpha = 180°$), the centrifugal force C will be equal to the force of gravity of the particle. Then we have

$$n_1 = 42.3/\sqrt{D} > n_{cr} \qquad (12.14)$$

In practice, the critical rotational speed is taken as

$$n = (0.50\text{-}0.65)\, n_1$$

The energy of pipe rotation is spent for overcoming the resistances of the support rollers and performing the work of lifting the material on the pipe walls. Assuming approximately that the pressure on the support rollers is the same at both sides, we can find the force of pressure as

$$2P = (G + G_0)/\cos \alpha_0 \qquad (12.15)$$

where G is the force of gravity of the load, G_0 is the force of gravity of the pipe, and α_0 is half the central angle of arrangement of support rollers (Fig. 12.9*b*).

If a is the distance from the mass centre of the material layer to the vertical line passing through the pipe axis ($a \approx 0.25D$), D_r is the diameter of the pipe rings which revolve on the support rollers, D_s and d are the diameters of the rollers and their axles, f is the

coefficient of sliding friction of roller axles, and k is the coefficient of rolling friction for the pipe rings on support rollers, the torque (Nm) on the pipe axis will be determined as

$$M = Ga + 2P \frac{2k + fd}{D_s} \frac{D_r}{2} \quad (12.16)$$

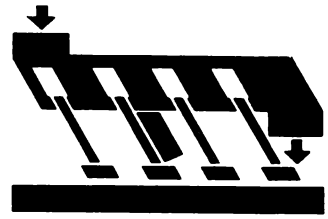
and the power of the drive motor, kW, will be

$$N = M\omega/1000\eta = M\pi n/30 \times \times 1000\eta = Mn/9586\eta \quad (12.17)$$

where ω is the angular frequency, 1/s, n is the rotational speed, rpm, and η is the efficiency of the drive transmission.

13

Oscillating Conveyers



13.1. Principal Types and Applications

An oscillating conveyer is essentially an open or closed trough or pipe which is elastically supported on a base structure or suspended from an overhead structure. A vibration exciter causes the trough or pipe to reciprocate (oscillate), so that the load in the trough performs successive short movements and propagates at a certain speed. The pattern of movements of the load depends on the movements of the trough which in turn are determined by the design and operating characteristics of the drive and the type of supports.

A strict and exhaustive classification of oscillating conveyers is hardly possible in view of the large diversity of their designs and close interrelations between them. By the pattern of motion of the trough and load, oscillating conveyers can be divided into two main types: (a) inertia-type, or rocking, conveyers (with a constant or variable pressure of the load on the trough bottom) and (b) vibrational, or shaking, conveyers.

In a rocking conveyer, the load moves relative to the trough due to inertia forces (the vertical component of the acceleration of the trough is smaller than the acceleration due to gravity). In a shaking conveyer, the load periodically rises above the trough ('jumps') and moves in small jerks (the vertical component of trough acceleration is greater than the acceleration due to gravity). This principal difference in the patterns of load motion determines different wear of the conveyer trough and participation of load mass in oscillating motion.

Rocking conveyers were employed widely for underground transportation of coal in collieries, but at present they are not used for this purpose in view of their inherent drawbacks. Shaking conveyers find various applications in the industry.

By the path configuration, oscillating conveyers can be divided into straight horizontal, slightly inclined (at an angle up to 15° upward or downward), and vertical (in which the load moves upwards along a helical trough).

Oscillating conveyers have the following advantages: relatively simple design; the possibility of perfect sealing of the conveyer for transportation of dusty, toxic and hot loads; the possibility for performing various process operations (screening, drying, cooling) together with conveying; little wear of the trough (in shaking conveyers) even by abrasive loads; and relatively low power consumption in steady operation.

The drawbacks of these conveyers include poor performance when conveying

loads along inclined (rising) paths; inapplicability with sticky loads; low conveying speed and low throughput capacity with fine-disperse pulverized loads; the transmission of vibrating foces onto the support structures, because of which the unbalanced masses of conveyers cannot be installed without foundation or on intermediate floors, and relatively short service life of the elastic elements and drive bearings. Besides, the troughs of rocking conveyers are subject to quick wear by abrasive loads.

Oscillating conveyers are employed for transportation of bulk loads and, less frequently, unit loads on horizontal or slightly inclined paths to relatively short distances (up to 50 m and rarely, up to 100 m) with throughput capacities up to 400 m^3/h. Vertical oscillating conveyers with a helical trough can raise bulk loads to heights up to 12 m and have throughput capacities up to 20 m^3/h.

The best application of oscillating conveyers (especially of the shaking type) is the transportation of dusty, hot, gas-evolving, toxic and chemically aggressive bulk loads in complete isolation from the surroundings in the chemical and metallurgical industries, manufacture of building materials, etc.

Horizontal oscillating conveyers are employed for transportation of iron and steel chips in machine shops, of hot knocked-out sand, foundry wastes and small castings in foundry shops, and hot articles (with a temperature up to 600°C) in metallurgical shops. Special types of vertical rocking conveyers are also used as collecting bins and feeders for delivery of small machine parts (screws, rivets, parts of instruments, etc.) in machining and assembly lines at machine-building and instrument-making plants and in automatic flow lines. Conveyers of this type are often called vibrating bins, or feed bins, though they operate, as a rule, as rocking, rather than as shaking (vibrating) conveyers, i.e. without shaking the load on the trough.

All the types of oscillating conveyers are inapplicable with sticky loads (such as wet clay) and poorly applicable with fine-disperse pulverized loads (cement etc.).

It is not advisable to employ oscillating conveyers (including the shaking types) for transportation of common bulk loads (such as sand, gravel, crushed stone, coal, grain) since they can be transported more efficiently by belt conveyers.

13.2. Dynamics of Oscillating Conveyers

If a plane inclined at an angle α to the horizontal is oscillated by an external power source with an acceleration j directed at an angle β (Fig. 13.1), a load particle on the plane will move along it. The normal pressure N of the particle on the plane (i.e. on the bottom of the trough or pipe of a conveyer) is the sum of the normal components of the gravity force and inertia force of the particle

$$N = mg \cos \alpha + mj_y \qquad (13.1)$$

where m is the mass of particle, g is the acceleration due to gravity, and

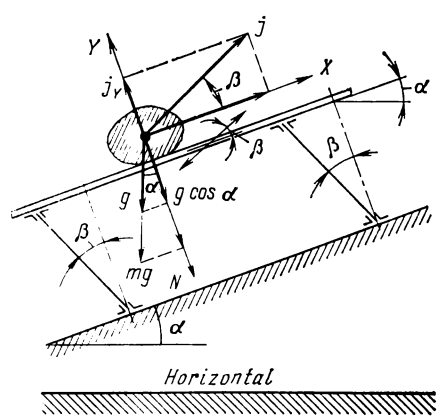


Fig. 13.1. Diagram to calculate the pressure of load on the trough

j_y is the component of the acceleration along the Y axis.

Let the X axis be directed parallel to the plane and the Y axis, normally to that plane, and the centre of coordinates be placed in the centre of gravity of the particle. Then the vertical (along the Y axis) component of the acceleration will be

$$j_y = -j \sin \beta = -a\omega^2 \sin \varphi \sin \beta \quad (13.2)$$

and the pressure of the particle on the plane

$$N = m (g \cos \alpha - a\omega^2 \sin \varphi \sin \beta) \quad (13.3)$$

where a is the amplitude of oscillations of the plane, ω is the angular velocity of the exciter, and $\varphi = \omega t$ is the phase angle of oscillations (here t is the time).

The relationship between the terms in brackets in the right-hand part of equation (13.3) characterizes the direction of the force of load pressure on the plane: with $g \cos \alpha$ greater than $a\omega^2 \sin \varphi \sin \beta$, it is directed downwards (let it be conditionally considered positive) and the load always remains in contact with the plane (trough); with $g \cos \alpha$ smaller than $a\omega^2 \sin \varphi \sin \beta$, the pressure force is directed upwards and the load tends to rise above the plane. The angle φ can vary between 0 and 360° and $\sin \varphi$ has the maximum value (equal to unity) at $\varphi = \pi/2 = 90°$.

The ratio of the highest normal component of the acceleration of the plane (trough), $j_{max} = a\omega^2 \sin (\pi/2) \sin \beta = a\omega^2 \sin \beta$, to the acceleration component due to gravity, $g \cos \alpha$, is called the coefficient (C) of operating mode of an oscillating conveyer

$$C = a\omega^2 \sin \beta / (g \cos \alpha) \quad (13.4)$$

For a horizontal conveyer, $\cos \alpha = \cos 0° = 1$ and the coefficient C is determined as

$$C_h = a\omega^2 \sin \beta / g \quad (13.5)$$

The coefficient C characterizes the dynamic conditions of operation of an oscillating conveyer (dynamic forces acting on the drive and other conveyer elements) and the pattern of motion of load particles: with $C < 1$, the load is always in contact with the oscillating plane and moves without rising above it (the operating mode of rocking conveyers); with $C > 1$, the load at some moments rises above the oscillating plane and moves mostly in microjumps (the operating mode of shaking conveyers, see Fig. 13.2); with $C = 1$, a change from one mode to the other takes place. If optimal values of the coefficient C are determined for a particular conveyer (especially when $C >> 1$), it is possible to find the optimal solution for the principal problems: to organize the conveying of the load at the highest speed and in appropriate coordination with motions of the oscillating plane (conveyer trough), with the lowest dynamic forces acting on the drive and other conveyer elements.

If the plane performs harmonic oscillations with the coefficient $C = 1$, the trajectories of load particles in the vertical plane will coincide with the direction of motion of the plane (Fig. 13.3). In the range $1 < C \leqslant 3.3$ the motions of a particle comprise a number of stages which follow one another during an oscillating period of the plane.

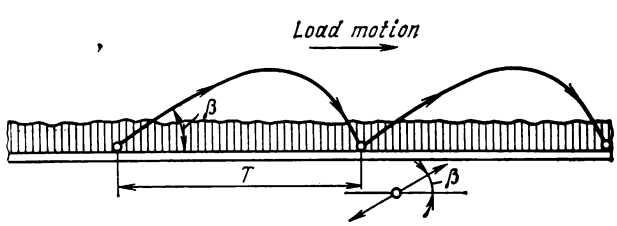


Fig. 13.2. Trajectories of particles on a shaking conveyer; arrows below indicate directions of trough vibrations

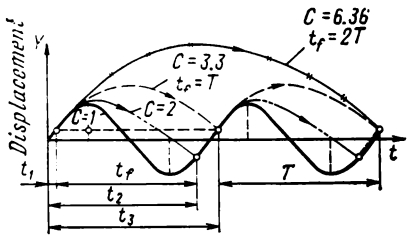


Fig. 13.3. Trajectories of material particles on shaking conveyer at various coefficients of operating mode C

For a certain time t_1, the particle moves together with the plane (trough). Then, as the phase angle increases and the coefficient C becomes greater than unity, the particle detaches from the oscillating plane and performs a free flight (microjump) during a time interval $t_f = t_2 - t_1$; after that, it falls back onto the plane and moves together with it during a time interval $t_3 - t_2$. The cycle of particle motion is then repeated.

The problem of finding the most efficient pattern of motion of load particles on a shaking conveyer (at $C > 1$) consists in correct selection of time t_2 (the moment when a particle falls onto the plane). The particle should be caught by the plane when being in forward motion and move together with the plane only for the shortest time needed to prepare it for a next jump; it should perform no sliding, intermixing, rearward or other motions which might interfere with its rational forward motion. From the standpoint of optimal displacements of the load, the best (optimal) modes of conveyer operation are those in which particles are tossed up incessantly and the time of their microflights t_f is equal to or multiple of the entire period of oscillations of the conveyer, T, i.e.

$$t_f = pT \qquad (13.6)$$

where p is an integer number.

As has been found experimentally, load particles can be tossed and move forward if

$$C = \sqrt{\pi^2 p + 1} \qquad (13.7)$$

With $p = 1$, i.e. when the time t_f of microjumps of load particles is equal to an entire period of conveyer oscillation, the coefficient $C = 3.3$; with $p = 2$, when $t_f = 2T$, we have: $C = 6.36$, and so on (see Fig. 13.3).

It has been established experimentally that, with $p > 1$ and $C > 3.3$, there are certain zones where the velocity of particles decreases. Besides, with $C > 3.3$, a conveyer must operate with substantial accelerations (say, greater than $6.6g$ at $\beta = 30°$), which involves high dynamic loads on the drive, bearings and other conveyer elements. Therefore, the coefficient of operating mode of shaking (vibrating) conveyers should be within the theoretical range $1 < C \leqslant 3.3$. The recommended values of the coefficients C, which can ensure the best conditions of load transportation and minimize dynamic loads, are given in Table 13.1 (according to the experimental results obtained by V. K. Dyachkov).

Table 13.1. Recommended Coefficients of Operating Mode (C) for Oscillating Conveyers

Type of conveyer	Type of vibrational drive	Coefficient C for loads	
		dusty and pulverized	lumpy
Single-trough or pipe, light- and medium-duty ($Q \leqslant 50$ t/h) of suspended or supported type	Centrifugal or electromagnetic	3-3.3	2.8-3
Ditto, heavy-duty ($Q > 50$ t/h)	Ditto	2-2.5	1.8-2.3
Single- and double-pipe, balanced, light- and medium-duty ($Q \leqslant 50$ t/h and $L \leqslant 30$ m)	Eccentric	1.6-2.8	1.5-2.5
Ditto, heavy-duty ($Q > 50$ t/h and $L > 30$ m)	Eccentric	1.3-2.5	1.2-2

13.3. Rocking Conveyers

In rocking (inertia-type) conveyers, the load is displaced under the action of inertia forces. By the mode of motion of the load-carrying element (trough) and the pressure of the load on the trough, rocking conveyers are divided into two main types: with a constant pressure (the systems of Marcus, Eickhoff, etc., the FRG) and with a varying pressure (the Kreiss system).

A Marcus conveyer consists of a trough supported by stationary rollers and a double-crank drive. The trough performs linear reciprocating motions in the horizontal plane, with its velocities and accelerations in the forward and return strokes being different. In an appreciable portion of the forward stroke, the trough velocity increases gradually, then drops sharply to zero and changes the direction. In the return stroke, the trough velocity first increases sharply and then smoothly decreases. The trough acceleration first increases gradually, then drops down sharply, changes the direction, attains the highest negative value and finally increases sharply. This mode of through motion is ensured by a double-crank or bar-link drive.

During a certain time interval in the forward stroke, the load moves together with the trough without sliding and accumulates kinetic energy. When the trough velocity changes sharply and the acceleration attains a certain highest limit, the load continues to move forwards by inertia with a certain deceleration and slides forward in this time interval and during the return motion of the trough. The latter slides off, as it were, from beneath the load. At a certain moment of the return stroke, the load ceases to slide on the trough. Thus, the load moves forward with a varying speed during the major portion of the forward and return strokes of the trough and moves rearward on the trough only for a very short time interval in the entire oscillating cycle (i.e. of each rotation of the crank). The operating mode of a conveyer is chosen so as to minimize the backward motion of the load. The force of pressure of the load on the trough bottom in each oscillating cycle is always the same and equal to its force of gravity; the force of friction of the load on the trough is also constant. All the time, the load lies on the trough and is not raised above it. This constant friction of the load causes wear of the trough.

Marcus conveyers have the following main characteristics: amplitude of trough oscillations, which is determined by the crank radius, 50-150 mm; frequency of oscillations (rotational speed of the drive shaft) 40-85 min^{-1}; trough width 200-1200 mm; trough length up to 50 m (sometimes up to 100 m); speed of load on a horizontal conveyer usually not more than 0.2 m/s.

A Kreiss conveyer has a trough supported on elastic stands which are rigidly fastened to the trough and a supporting frame. The stands are inclined at an angle of 20-30° from the vertical. The trough is oscillated by a common crank drive. The crank rotates at a constant angular speed and sets the trough into reciprocating motions of the same kind both in the forward and return strokes, in a direction perpendicular to the supporting stands. Since the stands are mounted at an angle, the trough is raised somewhat in a forward stroke and lowered, in a return stroke. The elastic stands are made of spring steel 60C2 or 55C2 and serve as energy accumulators: in a forward stroke of the trough, they are bent, store kinetic energy and convert it into the potential energy of elastic forces; in a return stroke, they straighten up and give up the stored energy to the trough.

The velocity and acceleration of the trough are directed perpendicular to the supporting stands. In the forward motion of the trough, the vertical component of velocity (and acceleration) is directed upward, perpendicular

to the trough bottom and in backward motion, it is directed downward. As the trough moves forward and rises on the stands, a load particle on it is acted upon by the vertical gravity force and the vertical component of inertia force (both directed downwards), the horizontal friction force, and the horizontal component of inertia force. The force of pressure on the trough bottom is the sum of the gravity force and the vertical component of inertia.

For a load particle to move forwards in the forward stroke of the trough, it is essential that the force of friction on the trough bottom be greater than the horizontal component of inertia. In a return stroke, i.e. when the trough moves back and downwards, the vertical component of inertia is directed upward and the gravity force, downward, so that the load pressure on the trough, which is the difference of these forces, and the force of friction of the load decrease. For the load to move forward when the trough moves backward, it is essential that the horizontal component of inertia be greater than the force of friction. This is quite possible, since the pressure of the load and the friction force have decreased to that moment. Thus, the pressure of the load on the trough bottom in the forward and return strokes turns out to be different, so that the load can propagate continuously along the trough.

For the load to move forward during the forward and return strokes of the trough, the load should be always in contact with the trough and its inertia force during deceleration of the trough should be greater than the force of friction of the load on the trough. The first condition is satisfied if the vertical component of trough acceleration is smaller than the acceleration due to gravity. For that reason, the coefficient C for rocking Kreiss conveyers (with a variable pressure on the trough) is taken less than unity, so that the load never rises above the trough, but slides on it with a certain variable velocity.

The second condition is satisfied by choosing an appropriate acceleration of the trough

$$j_{tr} = n^2a/90 > fg/(\cos\beta - f\sin\beta) \quad (13.8)$$

where n is the rotational speed of the crankshaft; a is the amplitude of oscillations (equal to the crank radius); f is the coefficient of friction of the load on the trough; and β is the angle of incline of supporting stands from the vertical.

Kreiss conveyers have the following main characteristics: amplitude of oscillations (determined by crank radius) 10-20 mm; oscillation frequency 300-400 min^{-1}; trough width 200-1000 mm; average speed of load in a horizontal conveyer 0.15-0.2 m/s; in inclined conveyers, the speed of load decreases substantially with increasing angle of incline which usually does not exceed 10-15°.

The Kreiss conveyer actually occupies an intermediate place between inertia-type (rocking) and vibrating (shaking) conveyers. In view of their drawbacks indicated earlier, rocking conveyers are not popular at present (mainly employed in processing machines).

13.4. Shaking Conveyers

13.4.1. Principal Types

There are many types of shaking (vibrational) conveyers which can be classified by various features.

By the direction of load motion, they can be divided into horizontal, slightly inclined, and vertical.

By the method of attachment of the load-carrying element (trough or pipe), it is distinguished between shaking conveyers on free elastic damping suspensions (a freely oscillating suspended system), and those on inclined guiding stands (a supported system with inclined elastic elements). In support-

ed-type conveyers, inclined elastic stands not only serve as the supports for the load-carrying element, but also cause this element (trough) to oscillate in desired directions.

By the number of simultaneously oscillating masses in the system, it may be distinguished between single-, double- and multiple-mass conveyers.

As regards the dynamic balancing, shaking conveyers can be either balanced or unbalanced.

By the number of load-carrying elements, shaking conveyers are divided into single-element types (with a single trough or pipe) and double-element types (with two troughs or pipes). Single-element conveyers may have a single or double load-carrying element; in the latter case, two elements, say, two parallel pipes, are rigidly connected with each other and oscillate as a united whole. Conveyers of this type are employed, for instance, for simultaneous transportation of various kinds of loads.

Shaking conveyers can also be classified by their applications. For instance, shaking conveyers, feeders and meters are mainly used for transportation of bulk loads; screening conveyers convey bulk loads and at the same time can screen them into various fractions or separate foundry wastes from moulding sand; this group includes vibro-shakers.

By the characteristics and adjustment of the elastic support elements (oscillating system), it is distinguished between shaking conveyers with a resonant, subresonant and superresonant system. In the resonant system (adjusted to resonance), the frequency of excitation force of a vibration exciter, ω, is roughly equal to base frequency ω_0 of natural oscillations of the elastic system of conveyer (in practice, it can be taken for steady operation that $0.85 < \omega/\omega_0 < 1.1$). In the subresonant system, the frequency of the exciter is substantially smaller than the frequency of natural oscillations of the conveyer ($\omega \ll \omega_0$). In superresonant systems, the frequency of the exciter is much higher than the frequency of natural oscillations of the conveyer ($\omega \gg \omega_0$).

The resonant elastic system ensures a low power consumption in steady conveyer operation, i.e. the conveyer may have a high throughput capacity, but requires high starting forces in view of the high stiffness of the elastic system. Resonant elastic systems have found preferable application in medium- and heavy-duty conveyers.

Superresonant systems possess a low stiffness and require lower forces at starting, but consume more power in steady operation of the conveyer; they can operate steadily for a long time at various changes of the load. The principal drawback of superresonant systems is that the stresses in the elastic elements increase substantially as the amplitude of oscillations increases for a short time during the passage through the region of resonance at starting and mainly at stopping of the conveyer (during the periods when the frequency of the exciting force varies and turns out to coincide with the frequency of natural oscillations of the system). A number of special devices have been developed to eliminate this effect. Further, since the elastic elements have a low stiffness, they deform substantially (depress) under the gravity force. Therefore, superresonant systems are employed mainly in light-duty conveyers of suspended or supported type. Subresonant systems have only a limited application.

The load-carrying element of a horizontal shaking conveyer performs, as a rule, linear (or sometimes circular or elliptical) symmetrical harmonic oscillations (Fig. 13.4) with a sinusoidal variation of exciting force. Conveyers with other patterns of periodic variation of the exciting force are also known; double-frequency (biharmonic) oscillations are also used, but such conveyers have found no wide application.

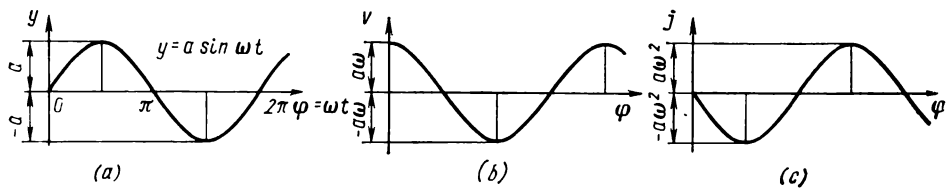


Fig. 13.4. Curves of various parameters of the trough of a shaking conveyer at harmonic oscillations
(*a*) displacement; (*b*) velocity; (*c*) acceleration

In vertical shaking conveyers, the load-carrying element performs double harmonic oscillations: linear along the vertical axis and rotational around that axis (i.e. longitudinal and torsional oscillations).

Oscillations of the load-carrying element can be excited by a centrifugal, eccentric (crank-and-rod), electromagnetic, hydraulic or pneumatic drive. Electromagnetic and eccentric exciters are most popular; hydraulic and pneumatic vibration exciters are used rarely, mainly for operation in explosive atmosphere. Shaking conveyers have rather low amplitudes of oscillations (usually 1-15 mm) and high frequencies (3000-400 min^{-1}).

Modern shaking conveyers should satisfy the following requirements: (1) the transmission of dynamic loads to the supporting structures should be minimized, so as to make the conveyer applicable on intermediate floors; (2) the possibilities should be provided for tight sealing of the transported material in the conveyers, in particular, at the end and intermediate points of loading and unloading; (3) it should be possible to load and unload the conveyer at intermediate points and automatically control these operations; (4) the total mass of the conveyer and the mass of oscillating elements should be as low as possible; (5) the conveyer should carry loads to an appreciable length without overloading, i.e. the optimal length of the conveyer path for a single drive; (6) the conveyer should not be high; (7) the elastic elements and drive should have a high reliability; the availability factor not less than 0.98 and the time to the first overhaul, not less than 8000 h; (8) the noise level of operating conveyer should be not more than 80-85 dB.

13.4.2. Effect of the Properties of Bulk Materials on Efficiency of Shaking Conveyers

In contrast to other types of conveyers, the efficiency of operation of shaking conveyers (throughput capacity, conveying speed, height of conveyed layer, etc.) depends substantially on the properties (mainly on particle size) of the transported material.

With the same oscillating parameters on a conveyer, various bulk materials will move with different speeds and form layers of different height in the trough, i.e. the conveyer throughput capacity will be different (Fig. 13.5). This feature of shaking conveyers should always be considered properly in their applications.

The best effect is obtained in transportation of dry uniform pulverized, granular and fine-size lumpy materials (sand, slag, coal, etc.); the speed of horizontal transportation of these materials reaches 0.3 m/s and sometimes, 0.6 m/s under the optimal oscillating conditions. Shaking conveyers are less efficient with non-uniform bulk loads containing a large percentage of dust particles; the speed of conveying of these materials is 50-66% lower. Dusty materials with the particle size less

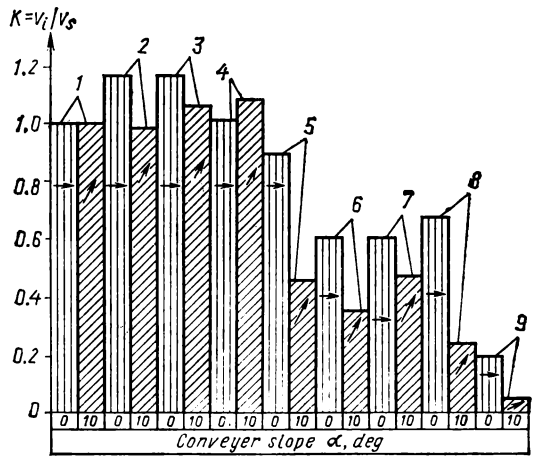


Fig. 13.5. Ratios of velocities of various materials v_l to the velocity of sand v_s in transportation on horizontal ($\alpha = 0$) and inclined ($\alpha = 10°$) shaking conveyers (amplitude of vibrations $a = 3.5$ mm, frequency 1000 min^{-1})

1—sand; *2*—small-size lumpy ore; *3*—small-size lumpy slag; *4*—lumpy coal; *5*—phosphate meal; *6*—apatite; *7*—powdered dolomite; *8*—powdered fireclay; *9*—cement

than 0.05 mm (such as cement) are even less suitable for transportation on shaking conveyers. The speed of their horizontal transportation is usually only 0.1-0.15 m/s and the layer height, 50-60 mm, even at the highest amplitude of oscillations.

As dusty particles 'jump', air interlayers form between them, so that the air resistance to flying particles increases. This effect explains why shaking conveyers are inefficient with dusty fine-disperse materials. An important problem is to find the optimal vibrating conditions and develop a shaking conveyer suitable for efficient transportation of dusty materials. With an increase of the oscillation amplitude, the efficiency of transportation of dusty materials increases. For that reason, such materials are mostly conveyed on shaking conveyers with an eccentric drive (oscillation amplitude 12-15 mm, frequency 500-400 min^{-1}). In conveyers with electromagnetic vibrators which have a low amplitude (0.5-1.2 mm) and high frequency (3000 min^{-1}), the transportation of dusty loads is practically impossible. In inclined conveyers with the load moving up the slope, the speed and throughput capacity Q diminish roughly by 3-5% per each degree of sloping angle (Fig. 13.6) and for that reason the angle of slope of shaking conveyers usually does not exceed 10°.

With pulverized loads, such as cement, and with ungraded materials containing more than 60% of dust particles, the angle of incline of conveyers

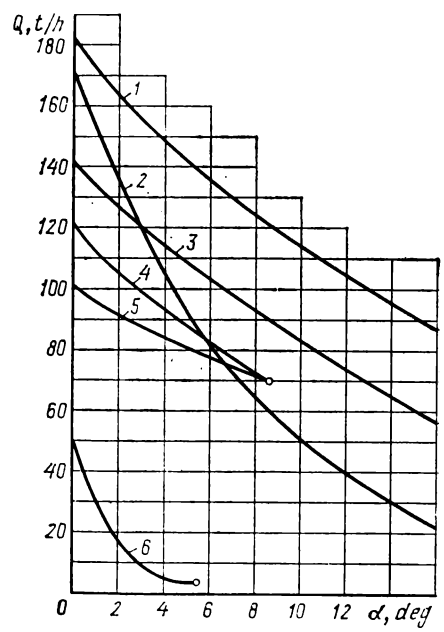


Fig. 13.6. Experimental curves of throughput capacity Q of double-pipe shaking conveyer (pipe diameter 310 mm, amplitude of vibrations 12.5 mm, frequency 600 min^{-1}, operating mode coefficient $C = 2.52$) at various angles of incline α

1—blast-furnace slag (lump size 0-35 mm); *2*—sand of natural moisture content; *3*—oil shale coke (lump size 0-30 mm); *4*—dry clinker (lump size 0-10 mm); *5*—crushed stone (size 0-30 mm); *6*—dry cement

is usually not more than 5°, since at higher angles the material practically ceases to move up to slope.

In declined conveyers (sloping downward at an angle up to 10°), all kinds of bulk materials are moved more easily. For that reason, the path for conveying of dusty bulk materials should be preferably declined at 10-5°.

A slight moisture content (up to 10 %) in non-sticky and non-viscous materials (clean sand, sawdust) does not impair their motion and sometimes is even favourable in increasing the speed; at a higher moisture content, however, the effectiveness of their transportation worsens sharply. Sticky wet cohesive materials (such as clay) can hardly be transported. Attempts are known to line the trough of shaking conveyers with various plastics or rubber for better transportation of sticky materials, but no noticeable success has been achieved.

Graded large- and medium-size lumpy materials, if they contain no fines, are readily transported on shaking conveyers, but produce much noise by striking the trough bottom and cause its wear; crushing of lumps is also possible. For higher wear resistance, the trough walls can be lined with rubber.

Ungraded materials with a wide range of particle sizes show low speeds of transportation. A transported material is not separated completely into fractions in a conveyer, though it can be observed that larger lumps move more quickly over the layer of finer particles.

In vibroshakers (racking screens) which have a perforated trough to screen off larger particles from finer ones, the speed of transportation is lower by 10-20 % than in common shaking conveyers with a smooth trough. The density of tran ported material has no noticeable effect on transportation speed; for instance, sawdust and iron ore are transported under comparable conditions at the same speed, though their densities differ substantially. Mixed loads, for instance, composed of layers of sand, chalk and crushed brick of the same particle size, are transported in a single flow without separation.

The speed of transportation of uniform lumpy loads is practically independent of the height of layer and remains constant with the coefficient of filling of the trough between 0.2 and 0.8. Easily moving dusty and dry granular materials are conveyed at the highest speed in a thin layer. In thicker layers, their speed decreases markedly, so that the height of the layer should be not more than 50-100 mm.

Thus, the properties of the materials to be transported should be considered properly in applications of shaking conveyers.

13.4.3. Horizontal and Slightly Inclined Shaking Conveyers

Suspended conveyers. A suspended shaking conveyer with a freely oscillating single-mass system (Fig. 13.7*a*) consists of a load-carrying element (pipe or trough) *1* which is freely suspended (or supported, Fig. 13.7*b*) on elastic ties—dampers *3*—from a stationary structure and is oscillated in a directed manner by a centrifugal drive *4*; the conveyer is provided with a safety belt *5* to hold the trough on occasional breakage of the elastic ties.

The drive may be in the form of a centrifugal (inertia-type unbalanced) vibration exciter of the pendulum type, two such exciters, or two synchronized electric motor-vibrators (Fig. 13.7*c*). The drive may be arranged below (solid lines in Fig. 13.7) or above (dotted lines) the load-carrying element. Synchronized motor-vibrators can also be arranged at the side of the trough (Fig. 13.7*c*).

For the load to move in the desired direction, the drive must be mounted so that the line of action of the excitation force is at an angle $\beta = 20\text{-}30°$ (direction angle) to the longitudinal axis of

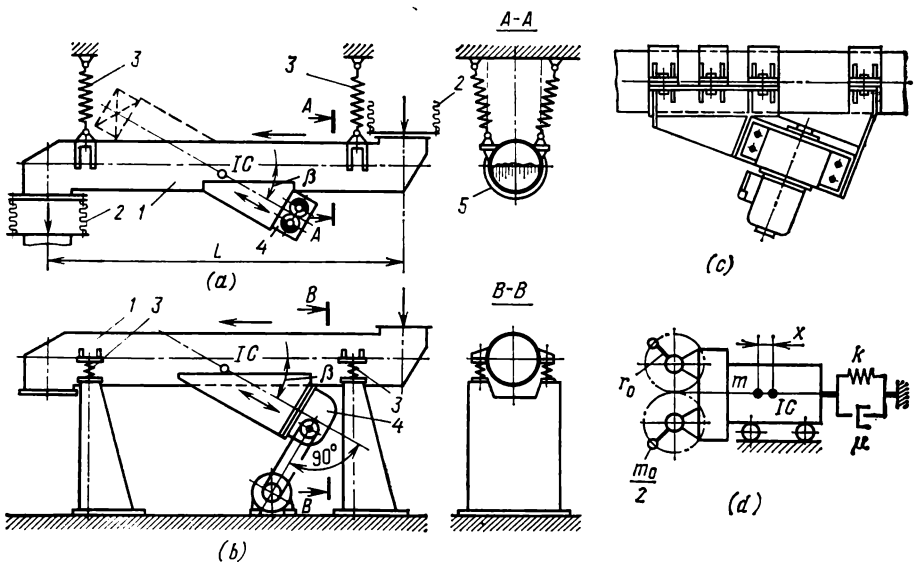


Fig. 13.7. Schemes of single-mass freely oscillating shaking conveyers with centrifugal vibration exciter
(*a*), (*b*), (*c*) design schemes; (*d*) calculation diagram; *IC*—inertia centre

the conveyer. Since load particles perform microjumps on the trough, the load is transported in the direction of the line of excitation force above the longitudinal axis of conveyer (or above the top level of the layer in a pipe), as illustrated in Fig. 13.7.

To induce strictly oriented linear oscillations of the load-carrying element, the conveyer drive should be arranged so that the line of excitation force (or the resultant of two forces from two vibrators) passes through the inertia centre (IC) of the entire oscillating system.

For a system completely symmetrical in the cross-sectional plane, the inertia centre coincides with its centre of gravity whose coordinates in space are determined by the well-known laws of analytical mechanics. If this condition is not observed, i.e. the line of excitation force does not pass through the centre of inertia (gravity) of the system, a moment of forces can form, and the system will additionally acquire torsional vibrations unfavourable for the process of load transportation.

For steady propagation of vibrations, it is essential to ensure a high stiffness of the joint of attachment of the drive to the load-carrying element of the conveyer. With a low stiffness, local oscillations of the walls of the trough or pipe can appear, which may have an unfavourable effect on load transportation and cause damage to the walls and the drive attachment.

In conveyers with a V-belt transmission from the electric motor to the shaft of a centrifugal drive, the axis of the belt transmission (Fig. 13.7*b*) should be perpendicular to the direction line of oscillations. Only in that case, vibrations of the conveyer will not be harmful for the transmission and the latter will not restrain the amplitude of vibrations of the conveyer trough (or pipe). Elastic suspensions (or supports) of shaking conveyers have a low stiffness and only serve as dampers; the frequency of natural oscillations of a system with elastic suspensions is much lower than the frequency of the excitation force. For that reason, conveyers with a freely oscillating system operate

in the superresonant mode which eliminates almost completely the transmission of vibration stresses from the conveyer to the supporting structures.

The loading and unloading connections of the vibrating pipe of a shaking conveyer are connected to stationary structures, such as bunkers or funnels, by means of flexible corrugated pipe connections *2* (Fig. 13.7*a*) made of strong fabric, rubber or plastics. These connections ensure proper tightness of the joints, but do not restrain the vibrations of the trough or pipe.

If a conveyer is mounted directly under the discharging hole of a bunker, the vertical pressure of the material column in the bunker should not be transmitted to the conveyer, but should be absorbed by the walls of the guiding funnel through which the conveyer is loaded. If this is impossible, a special feeder should be used to load the conveyer. These recommendations relate to all types of shaking conveyers.

The amplitude of vibrations of the load-carrying element of a freely vibrating conveyer can be determined by solving the differential equation of forced oscillations of the inertia centre of the system (Fig. 13.7*d*)

$$(m + m_0)\ddot{x} + \mu k\dot{x} + kx = m_0 r_0 \omega^2 \sin \omega t$$

where k is the stiffness of the elastic ties and μ is their internal resistance.

Neglecting the resistances in the elastic suspensions ($\mu = 0$) and making certain other assumptions admissible for a superresonant system, the amplitude of vibrations of a conveyer, a (cm), can be determined by the formula

$$a \approx \frac{m_0 r_0}{m} \qquad (13.9)$$

where m_0' is the total unbalanced mass of a centrifugal exciter; r_0 is the eccentricity of the unbalanced masses, i.e. the distance from their centre of gravity to the axis of rotation; and m is the total mass of vibrating elements of the conveyer including the attached

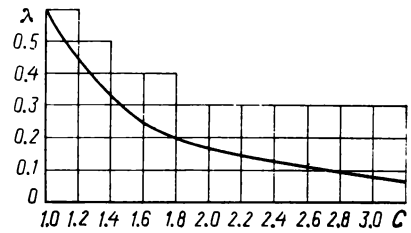


Fig. 13.8. Experimental curve of coefficient λ depending on the coefficient of operating mode C (according to V. K. Dyachkov)

mass of the load

$$m = m_{tr} + m_{ex} + \lambda m_l \qquad (13.10)$$

Here m_{tr} is the mass of the trough (pipe) with all the elements attached to it; m_{ex} is the mass of the vibration exciter connected with the trough; λ is an empirical coefficient which considers the mass of transported load and depends on the coefficient C of the conveyer operating mode (Fig. 13.8) and m_l is the mass of load in the trough (or pipe) at the rated coefficient of filling.

The length of a suspended-type shaking conveyer depends on the strength and stiffness of the load-carrying element and usually does not exceed 6 m, since a longer trough would have a heavier mass and smaller amplitude of vibrations.

The length of a shaking conveyer can be increased to a maximum of 8 m by providing a reinforcing rib along the pipe or trough (Fig. 13.9*a*).

The advantages of suspended-type shaking conveyers include a rather simple design, relatively low mass, the possibility of intermediate loading and unloading, and low dynamic forces acting on the supports. Among their drawbacks are a short length and small amplitude of vibrations which diminishes further with an increase of the load mass. Notwithstanding these drawbacks, suspended conveyers are used widely. Their main characteristics are as follows: pipe diameter 160, 200,

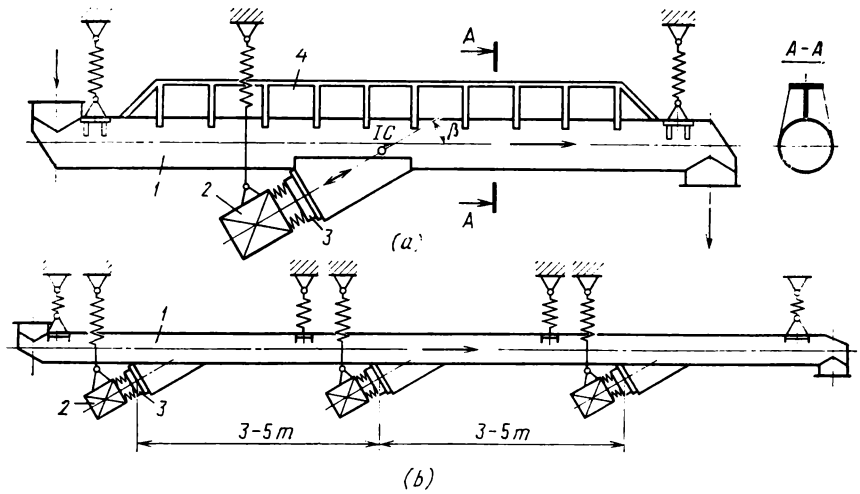


Fig. 13.9. Schemes of suspended shaking conveyers with electromagnetic drive
(*a*) double-mass; (*b*) multiple-mass

320 or 400 mm; throughput capacity 6-50 m³/h; and length 4-6 m.

A suspended shaking conveyer with an electromagnetic exciter represents a double-mass oscillating system (Fig. 13.9*a*); the first mass is formed by the load-carrying element (pipe or trough) *1* with the active portion of the electromagnetic exciter *3* which is attached to it, and the second mass is the reactive portion *2* of the exciter. All other elements are essentially the same as in a suspended conveyer with a centrifugal drive.

Suspended conveyers may employ single- or double-stroke electromagnetic vibration exciters. Multiple-drive (multiple-mass) suspended shaking conveyers are also known, in which two or four electromagnetic vibration exciters are attached to the load-carrying element (Fig. 13.9*b*). With several vibration exciters attached to the pipe (or trough), it is difficult to ensure their cophasing operation. For that reason, multiple-drive shaking conveyers (both suspended and supported) have only a limited application.

Supported shaking conveyers. A supported shaking conveyer with inclined elastic stands (Fig. 13.10*a*) comprises a load-carrying element *4*, elastic support elements *3* inclined at the direction angle β of vibrations relative to the vertical, a vibration drive *2*, and support frame *1* which can be placed directly on a foundation or on elastic dampers *5* (Fig. 13.10*b*). The vibration exciter may be centrifugal, electromagnetic (single-stroke) or eccentrical. Centrifugal and electromagnetic vibrators can be arranged above or below the conveyer. The excitation force of any type of drive must be directed at the direction angle β of oscillations and the line of its action must pass through the inertia centre of the oscillating system. In some designs of double-mass supported conveyers, a centrifugal drive is mounted on a heavy supporting frame in order to increase the amplitude of vibrations of the load-carrying element. Since the oscillation amplitudes in a closed oscillating system are inversely proportional to the oscillating masses, the heavy supporting frame will have a low amplitude and the load-carrying element, a high amplitude of vibrations.

All recommendations given earlier for suspended shaking conveyers are completely applicable to supported conveyers.

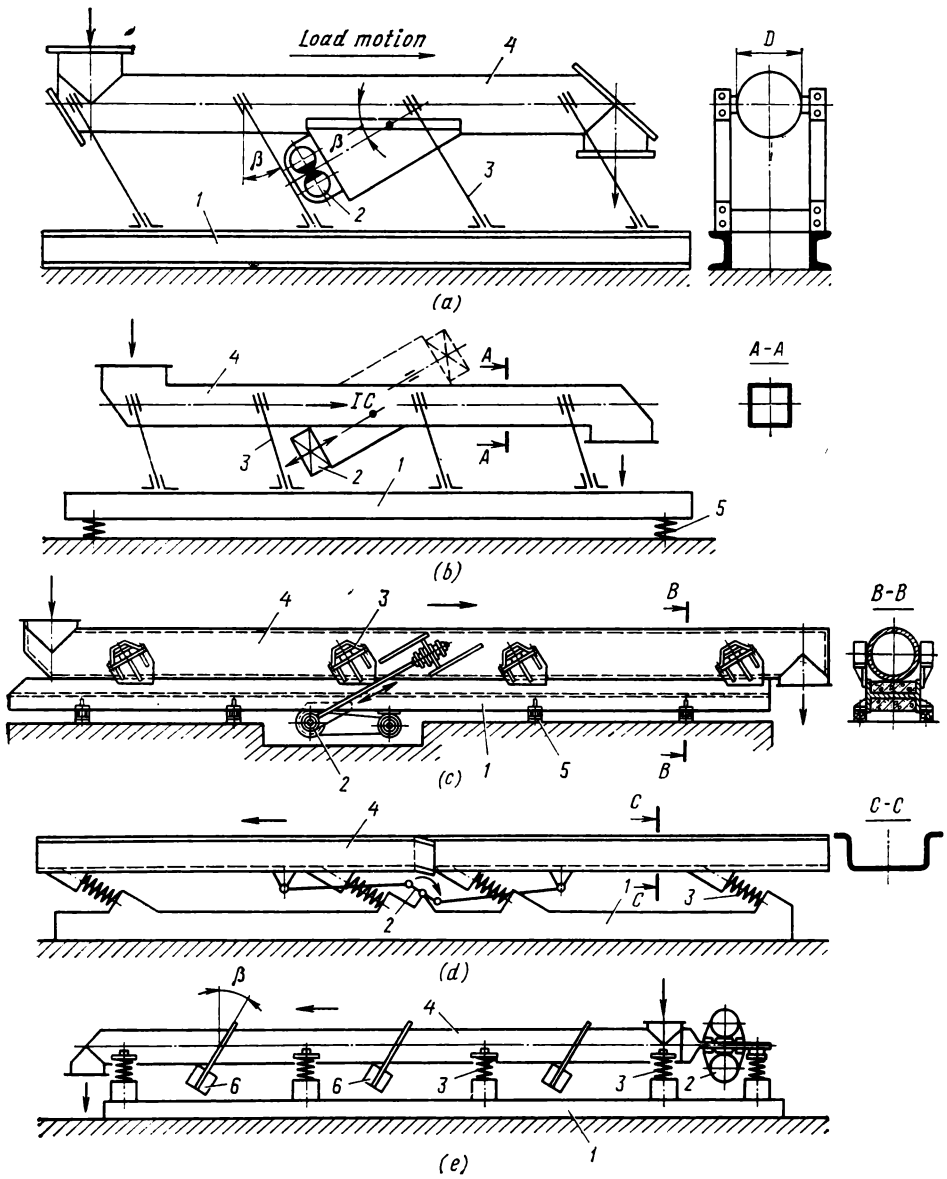


Fig. 13.10. Schemes of supported shaking conveyers
(*a*) single-mass; (*b*) double-mass with centrifugal (or electromagnetic) drive; (*c*) double-mass with a heavy supporting frame and eccentric drive; (*d*) double-mass doubled with eccentric drive; (*e*) multiple-mass with additional unbalanced masses

Supported conveyers with a centrifugal drive have a superresonant vibrating system, while those with electromagnetic and eccentric drives mostly have a resonant elastic system.

The main characteristics of supported shaking conveyers type КВЖГ with eccentric drive and elastic rubber system (Fig. 13.11) are given in Table 13.2. The trough in these conveyers

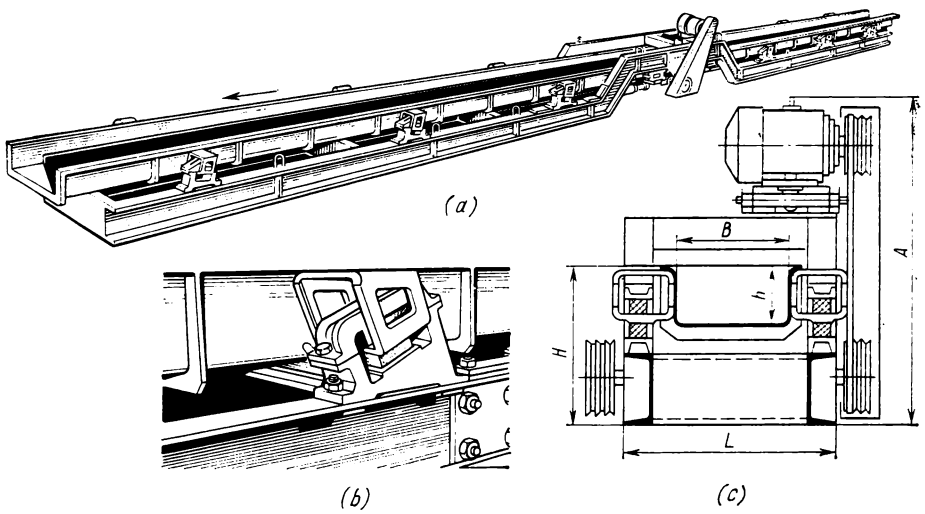


Fig. 13.11. Shaking conveyer type КВЖГ
(*a*) general view; (*b*) elastic rubber support; (*c*) cross-sectional view at the drive

oscillates with an amplitude of 6 mm and frequency 680 min^{-1}; the direction angle of vibrations is 22°; the transported material should have a temperature not more than 100°C.

The principal and essential drawback of supported shaking conveyers is their unbalance and the transmission of vibration stresses (mainly as reactions of elastic supporting elements) to the supporting structures, which requires a massive foundation. In attempts to diminish these stresses, some types of conveyers are provided with a heavy frame (which may be filled with concrete) whose mass exceeds 3-5 times the mass of the load-carrying element; the frame is mounted on elastic dampers (Fig. 13.10*c*). This design makes the conveyer excessively heavy.

Table 13.2. Characteristics of Conveyers, type КВЖГ (see Fig. 13.11*c*)

Characteristics	КВЖГ-200	КВЖГ-320	КВЖГ-500
Dimensions of trough, mm:			
width *B*	200	320	500
depth *h*	125	160	250
maximum length	24 000	30 000	38 000
Throughput capacity for sand in horizontal conveying, m^3/h	10	32	64
Electric motor power, kW	2.2/3.0	5.5/7.5	7.5/10.0
Linear mass, kg/m	80/96	130/150	210/240
Conveyer dimensions, mm:			
A	870	1050	1500
H	390	470	655
L	448	632	882

Note. Numbers in numerators relate to open-trough conveyers and those in denominators, to conveyers with tightly sealed trough.

In order to counterbalance the longitudinal forces, supported conveyers are sometimes made of two similar sections with the drive mounted between them (Fig. 13.10*d*). The connecting rods of the drive are connected with each section with a phase shift of 180°, which makes it possible to balance the horizontal components of the active forces; the vertical components remain unbalanced, and therefore, the conveyer must be installed on a foundation.

The supported conveyer manufactured by Uhde Co. (the FRG) has a load-carrying pipe (or trough) *4* (see Fig. 13.10*e*) mounted on elastic supports *3*, a centrifugal drive *2* consisting of two self-synchronizing motor-vibrators with unbalanced masses, a supporting frame *1*, and additional balancing masses *6* fastened on elastic brackets at the direction angle β of vibrations. Combined vibrations of the additional masses ensure directed motions of the pipe and load and improve somewhat the balance of the entire system.

The main characteristics of Uhde conveyers are as follows: pipe diameter 152-610 mm, length 12-27 m; linear mass 36-156 kg/m; and power of each motor 1-7.5 kW.

For supported shaking conveyers with a superresonant elastic system, centrifugal drive and a supporting frame rigidly fastened to the foundation (see Fig. 13.10*a*), the amplitude of vibrations can be determined by formula (13.9). For the conveyer illustrated in Fig. 13.10*b*, the formula gives only an approximate result. In conveyers with an eccentric drive, the amplitude a of vibrations is determined by the radius of the eccentric drive and is distributed between the frame and load-carrying element inversely proportional to their masses (see Fig. 13.12, $a = r/2$); if the frame is rigidly fastened on the foundation, the amplitude of the load-carrying element is equal to the radius of the eccentric ($a = r$). With drives having an elastic connecting rod (see later in the chapter), the amplitude of vibrations is smaller than the eccentric radius (Fig. 13.12*d*).

A double-pipe (double-trough) balanced shaking conveyer of supported type (Figs. 13.12 and 13.13) has two parallel load-carrying elements—pipes *7* and *3* (or troughs) connected together by hinged rockers *6* and elastic ties—leaf springs *5* (Fig. 13.12*a*) or rubber-metal elements *9* (Fig. 13.12*b*). The conveyer has an eccentric drive *8* mounted directly on the pipes (above or between them). Rockers *6* are attached hingedly on rubber-metal bushings to the pipes and support axles *4* which are carried by stands *2* connected with a stationary frame *1*. The rockers and springs are arranged at the direction angle β of vibrations. The connecting rods of the drive in the uppermost position are perpendicular to longitudinal axes of the springs and rockers and lie in the plane passing through the inertia centre (IC) of the system.

A double-pipe conveyer is essentially a balanced double-mass oscillating system in which the upper and lower pipe (or trough) serve as oscillating masses. Both pipes reciprocate under the action of an eccentric drive, parallel to each other with a phase shift of 180°, i.e. when one pipe moves forward, the other moves backward through the same distance, since the bearings of the drive shaft are connected with one of the pipes and the heads of the connecting rods, with the other. This ensures balancing of the moving masses of the conveyer. Since the pipes have the same mass, their amplitudes are also the same (see Fig. 13.12*c*).

The load moves forward in both pipes due to the inclined arrangement of the rockers. It will be transported in the direction of deflection of the longitudinal axis of rockers from the perpendicular drawn from the centre of the upper joint of rockers onto the longitudinal axis of the conveyer.

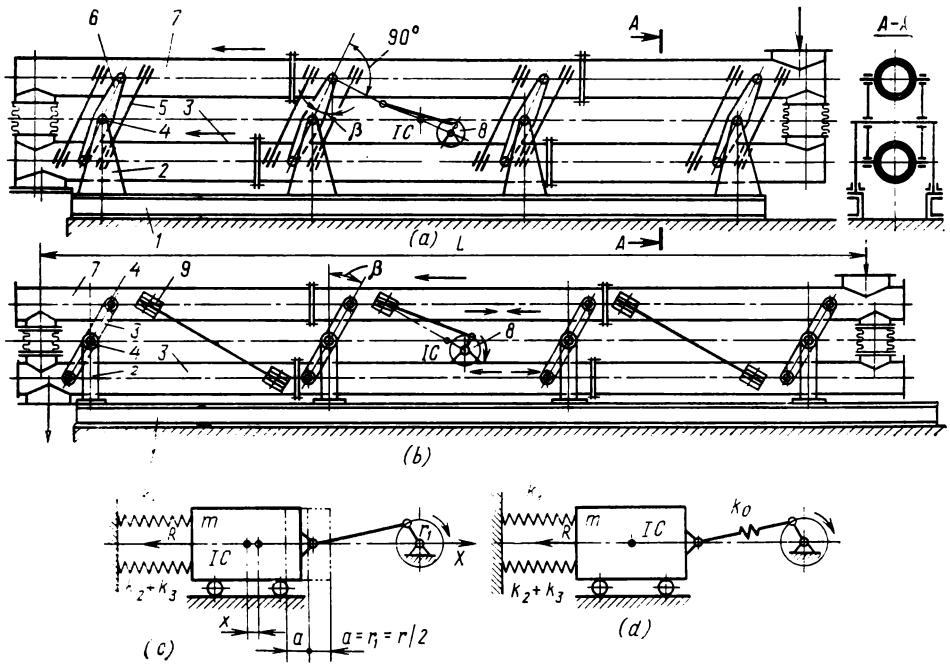


Fig. 13.12. Double-pipe balanced shaking conveyer
(*a*) with elastic spring system and rigid connecting rod; (*b*) with rubber-metal elastic system and elastic connecting rod; (c), (*d*) calculation diagrams for one vibrating mass of conveyers with a rigid and an elastic connecting rod; *a*—oscillation amplitude; *r*—eccentric radius

For the system to be balanced, it is essential to have the same masses of the upper and lower load-carrying elements and the same lengths of the upper and lower arms of all rockers; further, the longitudinal axis of the centres of all rockers should lie in the plane of longitudinal axis of symmetry of the load-carrying elements, which passes through the inertia centre of the entire system. A certain unbalance may appear if the pipes are loaded unevenly.

The main characteristics of double-pipe shaking conveyers with a rubber-metal elastic system (Fig. 13.13), manufactured by Binder Co., Austria, are given in Table 13.3. Their throughput capacity (for sand) ranges from 15 m³/h to 150 m³/h. The company also makes shaking conveyers with pipes and troughs of rectangular cross section. Conveyers with steel leaf springs are employed for transportation of hot loads.

Double-pipe shaking conveyers with pipes of a diameter of 160, 220, 320 and 400 mm for throughput capacities 10-70 m³/h and loads with a temperature of 500-850°C have been developed in this country.

A double-mass conveyer system in which the masses of the upper and low-

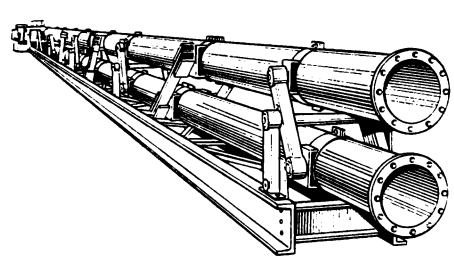


Fig. 13.13. Double-pipe shaking conveyer (Binder Co., Austria)

Table 13.3. Characteristics of Double-pipe Oscillating Conveyers Manufactured by Binder Co. (Austria)

Characteristics	Conveyer type					
	R10	R16	R22	R31	R39	R48
Pipe dimensions: inside diameter D, mm	104	154	220	310	390	470
maximum length, m	10	15	25	60	35	30
Cross-sectional dimensions of conveyer: A	250	390	700	750	900	1000
H	370	480	680	1100	1452	1680

er pipes and the displacements are fully symmetrical and equal, can be regarded approximately as a single-mass system of any of its halves possessing a single degree of freedom (see Fig. 13.12*c*). Then the differential equation of motion of the half-system with a rigid connecting rod will be of the form

$$m\ddot{x} + (k_1 + k_2 + k_3)\, x + R = P_0 \sin \omega t \quad (13.11)$$

where m is the mass of one load-carrying element of the conveyer with all elements fastened to it and with the attached mass of the load [see formula (13.10)]; x is the displacement of the inertia centre of the system along the X axis; k_1, k_2 and k_3 are the coefficients of stiffness of the springs and rubber-metal elements (k_1) and rubber-bushing joints (k_2, k_3) taken for a single mass; $R = i\gamma_0 S$ is the generalized force of resistance (here $i = \sqrt{-1}$ is the imaginary unit, γ_0 is the generalized coefficient of resistance; and S is the force of deformation of elastic ties); P_0 is the highest excitation force of the conveyer drive (at the maximum amplitude of the load-carrying element); and ω is the angular frequency of the excitation force.

The stiffness of elastic ties for a resonant system is chosen from the condition

$$0.85 < \omega/\omega_0 < 1.1 \quad (13.12)$$

where $\omega_0 = \sqrt{(k_1 + k_2 + k_3)/m}$ is the angular frequency of natural oscillations of the system.

Double-pipe conveyers can be loaded and unloaded at intermediate points through drum-type or tilting gates. Both pipes should be loaded evenly where possible.

The advantages offered by double-pipe shaking conveyers include the balanced arrangement of vibrating masses; doubled throughput capacity compared with single-pipe conveyers; a large transportation length attaining 60 m and even 100 m in a single-drive machine; constant amplitude of vibrations due to the application of eccentric drive; and low power consumption due to resonant adjustment of the elastic system. Among the drawbacks are a somewhat more complicated design of the conveyers and intermediate loading and unloading stations and large overall dimensions.

A balanced double-mass elastic system is also employed in conveyers where the lower load-carrying element is replaced by a counter-balancing beam of the same mass (counter-mass). Conveyers of this type are convenient when loading is carried out along the entire length of the conveyer or when transportation is combined with screening.

Conveyer Elements

The load-carrying element. The load-carrying elements of shaking conveyers are made in the form of pipes of circular, square or rectangular cross section or open or closed troughs of various shape. For dusty materials which are transported most efficiently in a thin layer, close troughs of rectangular cross section are employed.

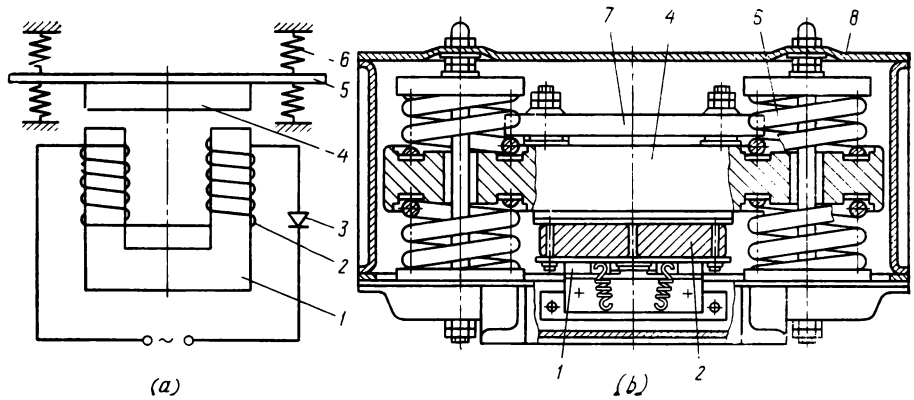


Fig. 13.14. Single-stroke electromagnetic vibration exciter
(*a*) schematic diagram; (*b*) design

Pipes and troughs are made from sheets of high-quality low-carbon or special steel, of a thickness of 3-5 mm and in sections 4-6 m long connected together by bolts with firm locks to prevent unscrewing. For higher wear resistance, pipes and troughs are lined inside with rubber, plastics (polyurethane) and like materials. Air cooling is employed when hot loads are conveyed.

The drive. The general concept of a vibration drive implies a combination of devices for excitation of mechanical oscillations, their transformation, and transmission to the working member of a machine.

The drive of shaking conveyers consists of a vibration exciter and electric motor, with or without an appropriate connection between them. It is also possible to employ pneumatic and hydraulic motors.

An electromagnetic vibrating drive has no electric motor per se.

An eccentric drive is a variety of the crank drive in which the crank is replaced by a disc mounted eccentrically on the drive shaft.

Shaking conveyers preferably employ electromagnetic, centrifugal and eccentric drives.

Electromagnetic vibration exciters may be of the single- or double-stroke type. A single-stroke electromagnetic exciter (Fig. 13.14) comprises a stator (electromagnet) *1* with windings *2* connected through a rectifier *3* to an a.c. power supply, an armature *4*, crosspiece *5*, elastic ties (springs) *6*, a set of governor weights *7*, and casing *8*. The cross-pieces and ties connect the stator with the armature. An elastic bond in the form of cylindrical springs or rubber elements determines the mutual position of the stator and armature, the gap between them, the kinematics of their linear reciprocal motion, and, together with the governor weights, the adjustment (usually resonant) of the conveyer oscillating system.

The stator and armature oscillate under the action of a pulsating force of magnetic attraction which appears in the stator when the stator windings are fed with a single-phase a.c. current. The attraction force is directly proportional to the current in the windings. With a sinusoidally varying current, the maximum force of attraction appears twice during a cycle, and therefore, the armature is attracted twice during the cycle to the stator and removed from it by the reactive force of springs. In that case, if the windings are fed with an a.c. current of a frequency of 50 Hz, the frequency of attraction of the armature to stator will be 100 Hz. Such a high frequency is of no use in shaking conveyers, be-

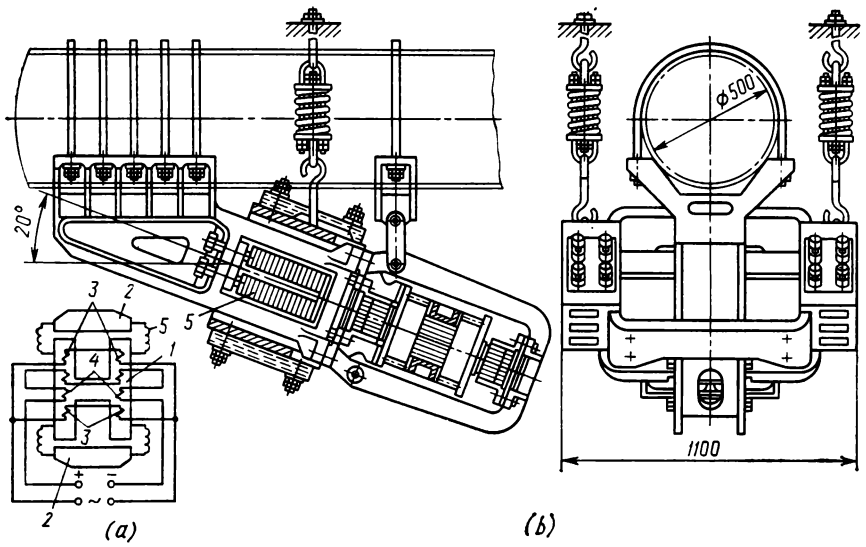


Fig. 13.15. Double-stroke electromagnetic vibration exciter
a) schematic diagram; (*b*) design

cause of which direct connection of stator windings to power supply is not employed. To diminish the frequency of oscillations of the vibrator, the stator windings are connected through a half-wave rectifier (selenium or valve type) which cuts off or decreases one of the half-waves of a sinusoidal current. As a result, the armature is attracted to the stator only once during a cycle of current variation, and therefore, the frequency of oscillations of the vibration exciter (and conveyer) will be equal to the current frequency, i.e. 50 Hz (or 3000 oscillations per minute).

The main advantages of single-stroke vibration exciters are small dimensions and low mass, and a drawback is a low power (usually up to 1 kW), so that they are used in conveyers and feeders of a low throughput capacity (roughly up to 20 m^3/h).

A double-stroke electromagnetic vibration exciter (Fig. 13.15*a*) comprises an H-shaped stator *1*, two armatures *2* rigidly connected with each other by a bracket, a.c. current windings *3* and d.c. windings *4*, and elastic ties *5* made in the form of a pack of leaf springs (Fig. 13.15*b*). A.c. and d.c. electric currents induce constant and variable magnetic fluxes in the stator which cause the armatures to reciprocate. When the upper armature is attracted to the stator, the lower one is repulsed, and vice versa.

Double-stroke vibration exciters are employed in double-mass feeders and heavy-duty suspended shaking conveyers; a serious drawback is that they have large dimensions and mass (up to 20 kg per 1 t/h of throughput capacity).

Feeders and conveyers with an electromagnetic drive, of the type described above, have the following characteristics: throughput capacity 50-650 t/h, vibrator power 0.5-8.0 kW, amplitude of vibrations 0.625-1.25 mm, and frequency 3000 min^{-1}.

Electromagnetic vibration exciters are provided with control means for smooth variation of the current in the windings of electromagnets which change the amplitude of oscillations, and therefore, the throughput capacity of a conveyer.

An advantage of electromagnetic vibration exciters is that they have no

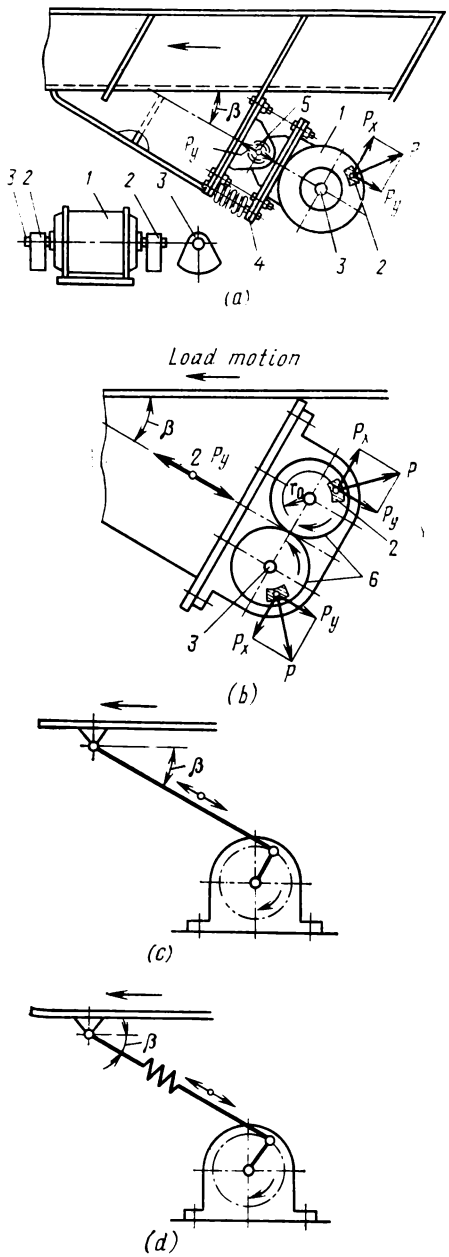


Fig. 13.16. Schemes of shaking conveyer drives
(*a*), (*b*) centrifugal; (*c*), (*d*) eccentric

sliding or rotating parts; further, they allow smooth control of throughput capacity and are easy in maintenance. Their drawbacks include an appreciable loss of throughput capacity on a voltage drop in the power supply circuit and a large mass of two-stroke exciters. A constant frequency (3000 min^{-1}) and small amplitudes (0.5-2 mm) restrict their application for transportation of pulverized bulk loads. They are mostly employed in feeders and conveyers of small length (up to 2.5-6 m).

Conveyer drives with centrifugal vibration exciters may be single unbalanced [with an elastic hinged (Fig. 13.16*a*) or rigid attachment and directed action of the excitation force], combined (Fig. 13.16*b*), or separate in the form of two unbalanced self-synchronizing motor-vibrators (see Fig. 13.7*c*). The last type is most popular at present.

An unbalanced single centrifugal vibration exciter (Fig. 13.16*a*) contains an electric motor *1* with an unbalanced mass (weight) *2* mounted on its shaft *3* eccentrically relative to the axis of rotation. In rotation, the unbalanced mass creates a centrifugal force

$$P = m_0 r_0 \omega^2 \qquad (13.13)$$

where m_0 is the unbalanced mass; r_0 is its eccentricity; and ω is the angular speed of the shaft.

The projections of the centrifugal force onto the coordinate axes are as follows

$$P_x = P \cos \varphi; \quad P_y = P \sin \varphi$$

The static moment of the unbalanced mass

$$M = m_0 r_0 \qquad (13.14)$$

In order to impart linear directed oscillations to the conveyer pipe, a single unbalanced drive (Fig. 13.16*a*) is provided with a supporting plate *4* and elastic joint *5*. The plate transmits through the joint to the conveyer only the longitudinal components of the centrifugal force, P_y, whereas the transverse components P_x are absorbed by the elastic joint (because of which the

drive is sometimes called the pendulum type) and are not transmitted to the conveyer. In a doubled centrifugal drive (Fig. 13.16*b*), two equal unbalancing masses *2* are mounted on two gear wheels *6* (or on two shafts *3*) which are engaged with each other. As the wheels rotate, centrifugal forces P appear; their longitudinal components P_y are added together (since they are directed in the same sense) and the transverse components P_x counterbalance each other (since they act in opposite directions). The highest directed (longitudinal) centrifugal force is determined by the formula

$$2P_y = 2m_0 r_0 \omega^2 \qquad (13.15)$$

Centrifugal drives find application in suspended and supported shaking conveyers and feeders.

Centrifugal drives are relatively simple in design, produce little noise in operation, can be sealed reliably, and afford a wide range of frequencies (2800-750 rpm) and excitation forces (up to 100 kN). A drawback is that their support bearings have a short service life (up to a year).

Eccentric (crank) drives with rigid (Fig. 13.16*c*), semi-rigid or elastic connecting rods (Figs. 13.16*d* and 13.17) are employed in single-pipe and mainly in balanced double-pipe shaking conveyers with a resonant elastic system. They have a wide range of amplitudes (up to 15 mm) and oscillation frequencies (from 400 to 800 per minute), but suffer from quick wear of support bearings.

In resonant-type shaking conveyers, the stiffness of the elastic system is extremely high, since it must accumulate a high kinetic energy of the pipe or trough with load in a forward stroke and to give it up in a return stroke. Therefore, a high force is required for initial deformation of such a stiff system at the conveyer start-up. In steady-state operation of the conveyer, the required energy is much lower, since it is spent only for overcoming the resistances in the elastic system and for displacement of the load.

High starting forces are coped with by the following methods: (1) the drive has a single motor of a high power sufficient to overcome the starting forces or with a high (for instance 10 times the rated value) starting torque; (2) the drive has two common motors which are used simultaneously at the conveyer start-up, after which (in steady operation) one of the motors is switched off; (3) an elastic element is added in the drive mechanism, which makes it possible to build up gradually the amplitude of the elastic system from a low value to the full working amplitude. The third method has found the widest application in modern conveyers as being the most efficient economically and effective. Elastic elements may be made in the form of semi-rigid or elastic connecting rods, elastic (rubber) eccentrics, a connecting rod with an oil damper, etc.

With an elastic element in the drive, the amplitude of vibrations of the pipe or trough is close in magnitude, but not equal to the radius of the eccentric, since both the main and the additional elastic ties participate in the operation of the oscillating system (see Fig. 13.12*d*).

A semi-rigid or elastic connecting rod is fastened to the load-carrying element through an elastic tie, i.e. a cylindrical spring or rubber-metal element (see Fig. 13.17). In a semi-rigid connecting rod, the elastic tie is restrained initially or has an appropriately selected stiffness, so that the rod operates as an elastic element at conveyer start-up and as a rigid element, in steady motion.

Elastic ties of the main oscillating system are made from steel leaf springs (Fig. 13.18*a*), cylindrical helical springs or rubber-metal elements (Fig. 13.18*c*). Elastic elements also include various rubber-metal joints (bushings, Fig. 13.18*b*) mounted in hinged joints of support rockers. Leaf and helical

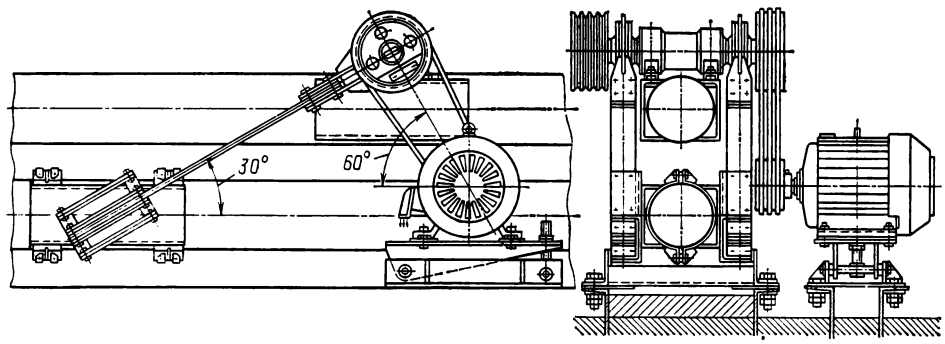


Fig. 13.17. Eccentric drive with elastic connecting rod for double-pipe shaking conveyers

springs are made of high-quality heat-treated spring steels 55C2, 60C2 or 60C2H2A; leaf springs are calculated for bending with a reduced allowable stress (up to 100-120 MPa) to ensure an elevated service life.

Springs are simple in design and manufacture, but develop high stresses at high amplitudes of vibrations and have a relatively low service life (1-2 years) because of high-frequency alternating stresses. Springs made of wood-laminated plastics and fibre glass plastics are also known. Pneumatic elastic ties may be promising in conveyers, but their application is limited by their intricate design.

Rubber-metal elements (Fig. 13.18*c*) have the best perspectives for application in shaking conveyers; they are shear-deformed and develop high amplitudes of oscillations and have a long service life. Unfortunately, they cannot operate at elevated temperatures (abo-

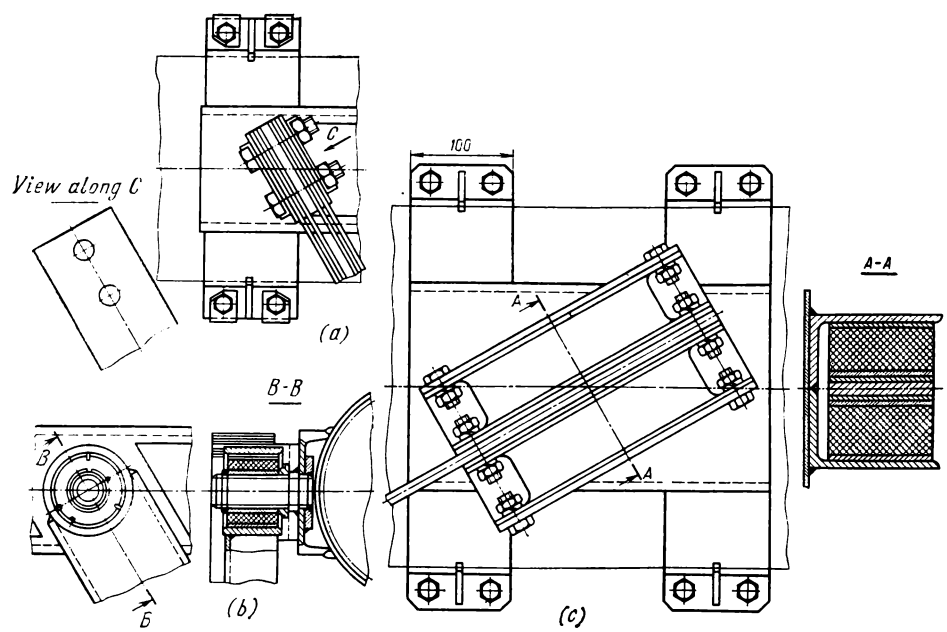


Fig. 13.18. Elastic elements of shaking conveyers
(*a*) leaf springs; (*b*) rubber-metal bushing (joint); (*c*) rubber-metal element

ve 100°C). Rubber-metal elements are, as a rule, used in elastic connecting rods.

Conveyer Calculation

The theory and calculation of a shaking conveyer involves three interrelated problems: (1) calculation of the oscillating system of a conveyer; (2) determination of the cross-sectional dimensions of the load-carrying element on the basis of the calculated average speed of the load and conveyer throughput capacity; and (3) determination of the power of the drive to overcome the inertia forces, resistances to load motion and losses in the elastic elements of the oscillating system.

Solution of the first problem gives the amplitude and frequency of vibrations, the forces in the drive and elastic ties, and their adjustment. The calculation is carried out along the principles of the theory of mechanical vibrations and the dynamics of oscillating systems with linear and non-linear ties, by compiling and solving the differential equations of motion of the system. The simplest equations of systems and their solutions have been given in descriptions of particular types of conveyers.

The frequency and amplitude of vibrations of a conveyer are determined by the recommended coefficient of operating mode (see Table 13.1) and can be taken from Table 13.4, depending on the type of drive and the characteristics of transported load. The values of frequency and amplitude thus found are substituted into the differential equation of motion of the system to determine the characteristics of the elastic ties of the conveyer and the parameters of the drive and the forces acting on it.

The kind of adjustment of elastic ties of the system (resonant or superresonant) is determined by the conveyer type as has been given earlier.

The direction angle of oscillations,

Table 13.4. Recommended Amplitudes a and Frequencies ω of Vibrations of Shaking Conveyers

Type of drive	ω, 1/min	a, mm, for loads	
		dusty and pulverized	lumpy
Electromagnetic	3000	1.2-2	0.75-1
Centrifugal single	2800-1500	1.2-3	0.8-2.5
Centrifugal double	1500-1000	2-4	2-3
Eccentric	800-450	5-15	4-8

β, is taken depending on the oscillation frequency ω: with $\omega \geqslant 1000$ min^{-1}, $\beta = 20\text{-}25°$; with $\omega < 1000$ min^{-1}, $\beta = 30\text{-}35°$; on the average, $\beta = 30°$.

The speed of load v (m/s) depends on the properties of the material and the angle of incline of the conveyer

$$v \approx (K_1 \mp K_2 \sin \alpha)\, a\omega \cos \beta \sqrt{1 - 1/C^2} \qquad (13.16)$$

where K_1 and K_2 are empirical coefficients depending on the physico-mechanical properties of the transported load (see Table 13.5); α is the angle of conveyer incline; a is the amplitude of oscillations; and C is the coefficient of operating mode of the conveyer (see Table 13.1).

In formula (13.16), the 'minus' sign in the brackets is taken for inclined

Table 13.5. Average Generalized Experimental Coefficients K_1 and K_2 according to V. K. Dyachkov)

Kind of load	Typical particle size, mm	Moisture content, %	K_1	K_2
Lumpy	5-200	—	0.9-1.1	15-2
Granular	0.5-5	0.5-10	0.8-1.0	1.6-2.5
Pulverized	0.1-0.5	0.5-5	0.4-0.5	1.8-3
Dusty	up to 0.1	0.5-5	0.2-0.5	2-5

conveyers (with the load moving up the slope) and the 'plus' sign, for descending conveyers (where the load moves down the slope).

For horizontal conveyers $K_2 \sin \alpha = 0$ and the speed

$$v = K_1 a\omega \cos\beta \sqrt{1 - 1/C^2} \quad (13.16')$$

Lower values of the coefficient K_1 and higher values of K_2 relate to loads with smaller particle size (finer particles can be conveyed at a lower speed, see Table 13.5).

The throughput capacity of a shaking conveyer is found by formula (2.8), with the coefficient of filling taken within the following limits: $\psi = 0.6$-0.9 for open troughs, $\psi = 0.6$-0.8 for rectangular pipes, and $\psi = 0.5$-0.6 for circular pipes; lower values relate to finer materials.

The width of a trough or diameter of a pipe should be checked for the largest size of lumps of the material. The largest lumps of ungraded loads should measure not more than 1/4 and those of graded loads, not more than 1/3 of the pipe diameter or trough width.

The power N (kW) of a drive motor with an elevated starting torque is determined by the empirical formulae proposed by V. K. Dyachkov:

for short conveyers (of a length up to 10 m)

$$N \approx \frac{C_t Q}{10^3 \eta} \left(K_3 L + \frac{H}{0.36} \right) \quad (13.17)$$

for conveyers of a length $L > 10$ m

$$N \approx \frac{C_t Q}{10^3 \eta} \left[10K_3 + (L - 10) K_4 + \frac{H}{0.36} \right] \quad (13.18)$$

where C_t is the coefficient of transportability of the load; for granular and lumpy loads (sand, coal, slag, grain) which possess a high transportability, $C_t = 1$; for pulverized and dusty loads of a reduced transportability (cement, cinder, apatites), $C_t = 1.5$-2; Q is the rated throughput capacity of conveyer, t/h; K_3 and K_4 are the coefficients of unit power consumption (Table 13.6); L is the horizontal projection of the length of load transportation, m; H is the lifting height (in inclined and declined conveyers), m; and $\eta = 0.95$-0.97 is the efficiency of the drive mechanisms.

13.4.4. Vertical Shaking Conveyers

A vertical shaking conveyer-elevator (Fig. 13.19*a*) has a rigid cylindrical casing (pipe) *1*; a helical open trough (or closed pipe) is attached at the out-

Table 13.6. Average Values of Coefficients K_3 and K_4

Conveyer type	Rated throughput capacity, t/h	K_3	K_4
Single-mass suspended with centrifugal drive	5-50	6-7	—
	above 50	5-5.5	—
Single-mass supported with inclined guide springs and centrifugal drive	5-50	7-10	5-6
	above 50	5-6	3.5-4
Single-and double-pipe double-mass balanced, with eccentric drive	5-50	with rigid connecting rods	
		10-12	8-10
	5-50	with elastic connecting rods	
	above 50	4.5-5	3.5-4
		4-5	3-3.5

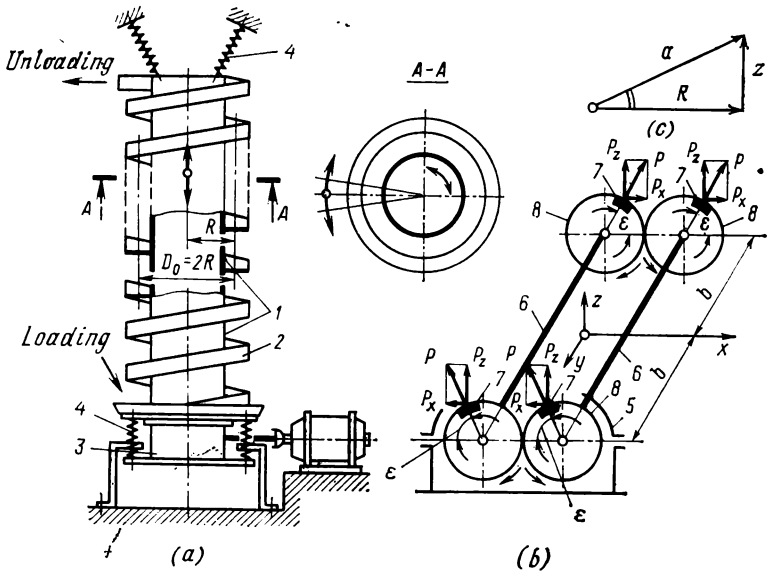


Fig. 13.19. Vertical shaking conveyer
(*a*) design scheme; (*b*) scheme of two-shaft centrifugal vibration exciter; (c) components of oscillation amplitude

side (or sometimes at the inside) of the vertical tubular casing, along which the load can be transported from the bottom upwards. A vibration-exciting drive *3* is mounted at the top or bottom of the casing to impart to the latter directed vibrations along and around the vertical axis (i.e. longitudinal and torsional oscillations), which cause the load to move upwards along the helix.

In a freely suspended version, the conveyer casing is suspended from overhead structures of the building or fastened at the top and bottom to a foundation by means of damping springs *4* or rubber dampers of a low stiffness, to prevent transmission of vibration stresses to the supporting structures (superresonant adjustment of elastic ties).

Vibrations in a vertical conveyer can be induced by two electromagnetic vibration exciters or unbalanced motor-vibrators mounted at the direction angle of oscillations on the conveyer casing; a single two-shaft centrifugal vibration exciter or an eccentric drive can also be used.

A two-shaft vibration exciter (Fig. 13.19*b*) has a strong housing *5* with two parallel shafts *6* arranged inside; the ends of the shafts carry discs with equal unbalanced masses *7* which are arranged pairwise at a definite angle to each other. The shafts are interconnected by cylindrical gear wheels *8* and rotate strictly synchronously and in phase from an electric motor. The rotating unbalanced masses develop centrifugal forces P. The vertical components of these forces, P_z, induce vertical vibrations in the conveyer (along the vertical axis); the horizontal components P_x are directed differently and form a moment which causes torsional vibrations of the conveyer. With a particular combination of these oscillations at definite frequency and amplitude, the load moves upwards along the helix.

Vertical conveyers with electromagnetic vibration exciters have a frequency of 3000 min^{-1}, those with centrifugal exciters, 1000-1500 min^{-1}, and those with eccentric drives, 600-700 min^{-1}. The total (directed) amplit-

ude of oscillations is usually $a = 0.5$-8 mm (see Fig. 13.18*c*).

For a given amplitude of oscillations a (cm), the mass m_0' (kg) of one of the four unbalanced masses of a two-shaft vibration exciter can be found by the formula:

$$m_0' = \frac{am}{4r_0}\sqrt{\left(\frac{bR}{\rho_i^2}\right)^2 \cos^2 \varepsilon + \sin^2 \varepsilon} \quad (13.19)$$

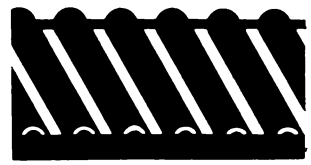
where m is the total mass of oscillating parts of the conveyer with the attached mass of the load, kg; b and R are the dimensions (cm) according to Fig. 13.18; $\varepsilon = 45°$ is the adjusting angle of unbalanced masses; r_0 is the eccentricity of an unbalanced mass, cm; and ρ_i is the radius of inertia of the conveyer, cm.

The throughput capacity of a conveyer is determined by formula (2.8); the speed of transportation can be found by formula (13.16) on substituting the helix angle ($\Omega = 4$-8°) for the angle of incline α. The outside diameter of conveyer casing is determined as $D_c \geqslant H/10$, where H is the lifting height. In existing vertical conveyers, $D_c = 300$-900 mm, trough width 100-400 mm, throughput capacity up to 20 m³/h, and lifting height 6-12 m.

A vertical shaking conveyer developed at the VNIIPTMash Institute has an eccentric drive and single-mass resonant vibrating system. The conveyer characteristics are as follows: the highest throughput capacity for sand 10 m³/h; outside diameter of conveyer casing 800 mm; trough width 200 mm; helix lead 200 mm, maximum height 9.7 m; vibration amplitude 6.5 mm; frequency 600-700 min^{-1}; the highest speed of transportation of sand 0.15 m/s; angle of arrangement of elastic elements 45°; power of electric motors for the largest lifting height, 2×4 kW; and linear mass of a load-carrying section 120 kg/m.

14

Roller Conveyers



14.1. Principal Types

Roller conveyers are employed for transportation of various unit loads (ingots, plates, moulds, rolled metal sections, tubes, boards, bottomplates, containers, boxes, etc.) on horizontal or slightly inclined paths. To roll on the conveyer rollers, the load units should have a flat or longitudinally ribbed bottom surface, a cylindrical shape, etc. The transported loads move on stationary rollers mounted in the frame of a conveyer.

By the method of operation, roller conveyers are divided into powered and unpowered. In the former, the rollers are driven from a motor and impart motion to the load units on them. In unpowered roller conveyers, the driving force is applied directly to loads and as these move, they cause the rollers to rotate.

Unpowered roller conveyers are mostly installed with a slight slope in the direction of motion, so that the loads are moved by the longitudinal component of their gravity force. Conveyers of this type are called gravity conveyers and represent essentially an inclined plane with the continuous floor being replaced by a set of rollers which diminish the resistance to motion, and therefore, may have a smaller angle of decline. On horizontal or inclined (sloping upward) unpowered roller conveyers, the loads are moved by an external force, say, a chain clamp, push bar, etc. In the simplest cases, light loads can be moved manually.

The spacing between rollers should be such that a load unit always lies on at least two rollers; practically, for smooth travelling of loads, this spacing is taken not more than 1/3 of the length of load units and even smaller for long load units which might sag between rollers.

The rollers of roller conveyers may be cylindrical or disc-shaped.

14.2. Unpowered Roller Conveyers

14.2.1. Types and Designs

Roller conveyers are divided into stationary or portable (Fig. 14.1*a*) and mobile, which are mounted on a wheeled undercarriage (Fig. 14.1*b*). A conveyer is assembled from individual sections. A section comprises a frame supported by stands and a set of rollers mounted in it. The stands may be made extensible to vary the angle of incline of the conveyer (Fig. 14.1*a*).

Rollers usually have a cylindrical shape and ball bearings to rotate on stationary axles. Unit loads of a cylindrical shape, such as tubes, round steel

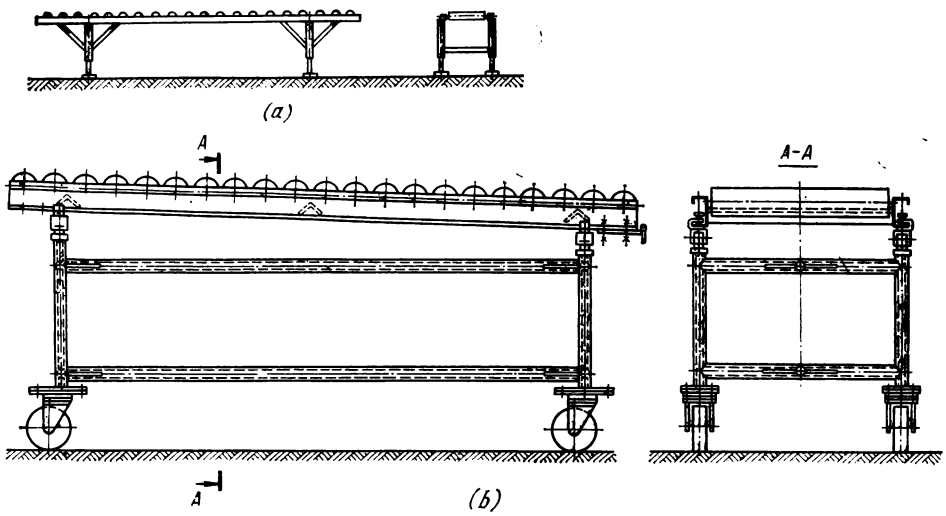


Fig. 14.1. Sections of unpowered roller conveyers

sections, etc. are transported on roller conveyers with tapered rollers or with roller pairs forming a trough. Roller pairs are also used in cases when loads are moved by a chain with clamps; the chain is arranged along the centre line of a conveyer between rollers in roller pairs. Roller bodies are mostly made from tube sections. Rollers made of plastics and other synthetic materials can be employed in light-duty conveyers. They have a lower mass, appropriate acid resistance, possess a certain elasticity, and produce much less noise.

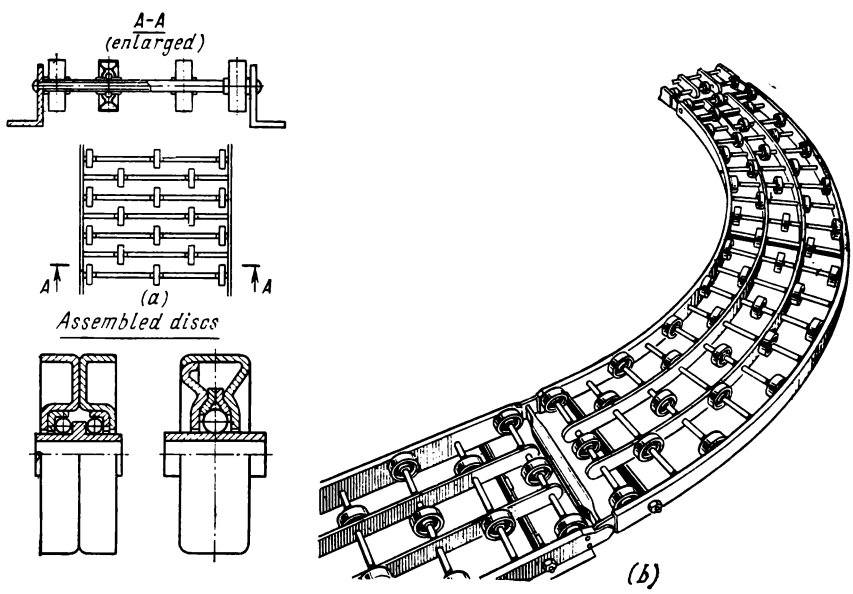


Fig. 14.2. Disc-roller conveyer
(a) straight-path; (b) curved-path

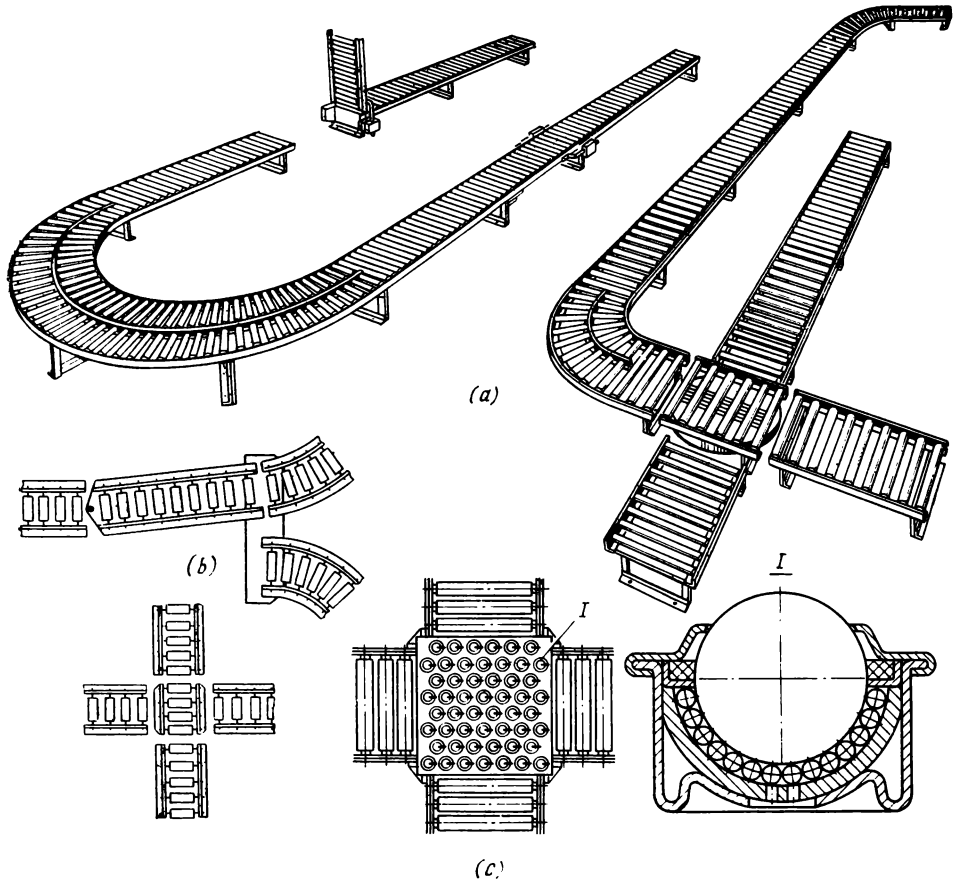


Fig. 14.3. Unpowered roller conveyers
(*a*) with curved sections; (*b*) switch; (*c*) crossing of two roller conveyers with a ball-type turntable

In many cases, disc-shaped rollers are employed instead of cylindrical ones. They are mounted on special ball bearings on stationary axles. Disc-shaped rollers may be of diverse design, for instance, with internal and external plastic rings, with single- or double-row bearings, with a steel axle and ring spacers (Fig. 14.2), etc. Disc rollers have certain advantages over the cylindrical type, for instance, in horizontally curved conveyer sections where roller discs may rotate at different angular speed (higher at the external side of a curve and lower at the internal side), so that unit loads move on them without slippage. In conveyers for small-size unit loads, the discs may be staggered and thus arranged closer for smoother travel of loads.

Unpowered roller conveyers may have various crossings and sideways with special transfer sections functioning like switches of rail tracks. In crossings, a turntable is provided which can be turned in-line with one or the other roller conveyer (Fig. 14.3*a* and *b*). Self-adjustable disc rollers on swivel heads and spherical (ball) supports are also in use, which allow motion of loads in any direction (Fig. 14.3*c*, *I*). In curved sections, cylindrical rollers are made up of a number of shorter sections or replaced by tapered rollers.

Figure 14.4 shows a distributing roller conveyer with four transverse re-

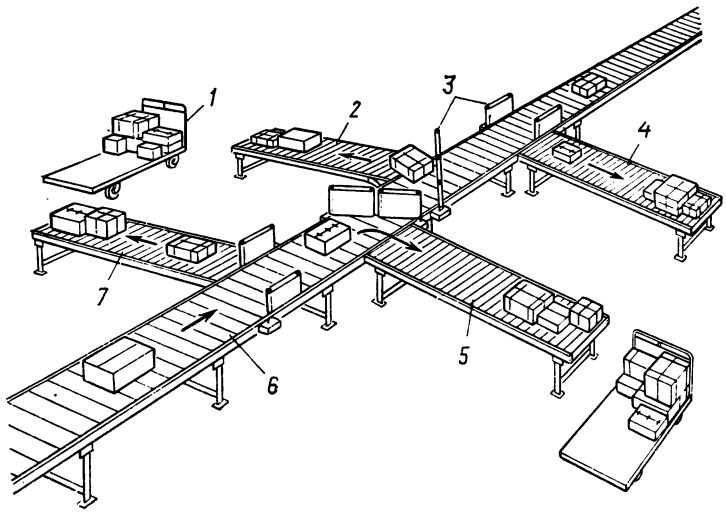


Fig. 14.4. Distributing unpowered roller conveyer with four receiving tables

1—electric car; *2*,*4*, *5*, *7*—receiving tables; *3*—shield controls; *6*—distributing conveyer

ceiving conveyer sections at its sides. Load units can be transferred from the main conveyer onto the receiving conveyers by automatically or manually controlled belt unloaders and are then reloaded onto electric cars.

An example of automatic addressing of container loads to sideway roller conveyers is illustrated in Fig. 14.5*a*. Each container has in its bottom a number of metal plates according to the number of sideways of the conveyer. A coded address is written magnetically on one of the plates. Readers are arranged before the sideways under the conveyer rollers. If the address coded on a container coincides with that set on a reader, the latter sends a command to a switch (a tilting lever) which transfers the container onto the sideway. If the coded addresses do not coincide, the container moves unobstructedly further on the main conveyer.

Roller conveyers arranged at a sharp angle to each other are provided with mechanical or electrical locking mecha-

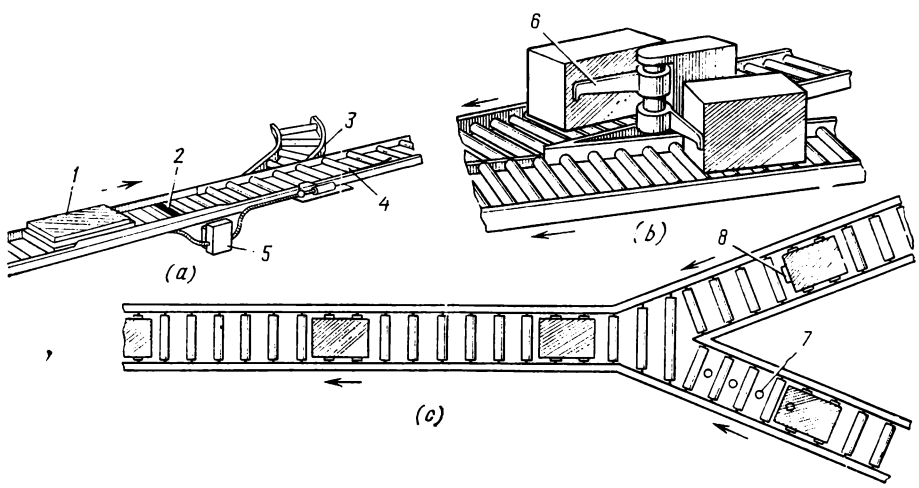


Fig. 14.5. Automatic switches

(*a*) load transfer onto a sideway conveyer; (*b*) and (c) locking devices; *1*—container; *2*—reader sensor; *3*—sideway conveyer; *4*—transfer lever; *5*—reader; *6*—stand with two-arm lever; *7*—electric switches; *8*—stop

nisms to prevent collisions of load units on them. A mechanical device with a two-arm lever is shown in Fig. 14.5b. When a load unit is moving on one conveyer, the lever does not allow other load units to move on the other conveyer. In an electric locking system (Fig. 14.5c), conveyers are provided with stops which are actuated from electric switches mounted in proper places of the conveyer path. When load units are moving along one of the conveyers, they engage a switch which moves a stop on the other conveyer into the locking position.

The radial run-out (wobbling) or rollers may range between 0.2 mm and 0.5 mm depending on their diameter, and the highest deviation from the plane should be not more than 0.4-1.2 mm. The length of cylindrical rollers is chosen from the standard series: 160, 200, 250, 320, 400, 500, 650, 800, 1000 and 1200 mm, and their pitch, from the standard series: 50, 60, 80, 100, 125, 160, 200, 250. 315, 400, 500, and 630 mm.

The rated load P on a roller is taken approximately depending on the ratio of the length of load units to the roller pitch, l_l/l_r; it is smaller at a higher ratio, since the gravity force of a load unit G_l is distributed a on larger number of rollers. With $2\ l_r < l_l < 3l_r$, it may be taken that $P = 0.5G$; with $3l_r < l_l < 4l_r$, $P = 0.33G$, and with $4l_r < l_l < 5l_r$, $P = 0.25G$.

14.2.2. Calculation of Unpowered Roller Conveyers

The motion of loads on stationary rollers of a roller conveyer is essentially the motion 'on wheels' which has been discussed in Sec. 2.2; it involves rolling friction of loads on rollers and sliding friction in roller bearings. In the general case, the coefficient of resistance of loads moving on rollers can be determined by formula (2.23).

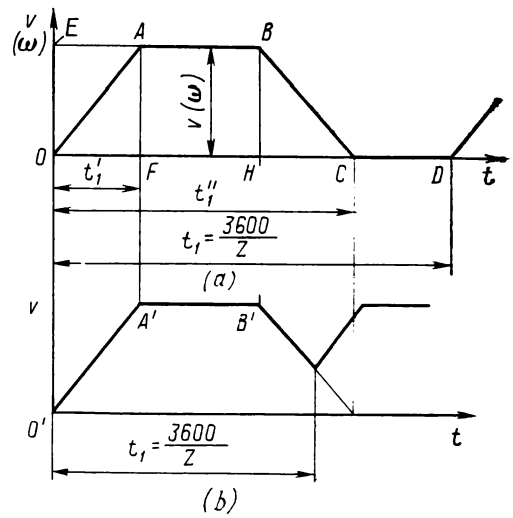


Fig. 14.6. Diagram of roller velocities on an unpowered roller conveyer

Let us calculate the force W required to move a load on a horizontal unpowered roller conveyer and the required angle of decline β of a gravity-type conveyer to ensure the load motion at a speed v (m/s).

As has been indicated, the conveyer rollers are rotated by the loads which move on them. After a load unit has passed a roller, the latter continues to rotate deceleratedly and may even stop fully before a next load runs onto it.

The diagram of velocities at the roller circumference, and therefore, of angular velocities of the roller for the case considered is shown in Fig. 14.6a. At the initial period t'_1, when the circumferential velocity of the roller is still smaller than the translational velocity v of a load unit (this velocity can be taken to be constant), the load rolls and slides on the roller. The velocity of the roller which is acted upon by a constant force of friction of the running-on load, increases by a linear law (OA in Fig. 14.6a). After that, the roller rotates at a constant circumferential velocity v (line AB). As the load runs off the roller, the latter begins to rotate deceleratedly (line BC) and then re-

mains at rest (line CD) until a next load runs onto it. The entire cycle is then repeated. With z load units being transported per hour, the time (s) of a single cycle $t_1 = 3600/z$.

Let G' be the pressure of the load on a roller and μ_0, the coefficient of sliding friction between them in motion. Then the friction force between them during acceleration of the roller is $G'\mu_0$ and the path of action of that force, vt_1' (the area $OEAF$ in Fig. 14.6*a*). Therefore, the work performed by the load is $G'\mu_0 vt_1'$. The path of any point on the roller circumference during that period is $vt_1'/2$ (the area OAF). Hence the path of sliding is also equal to $vt_1'/2$ (the area OEA). Thus, half of the total work of the load during the accelerated rotation of the roller is transformed into the work of friction and another half, into the kinetic energy of the roller. In other words, the work of the load is twice the kinetic energy of the roller, $2A$. For a rotating roller having the gravity force P, we can determine A by the formula:

$$A = KPv^2/(2g)$$

where $K < 1$ is a coefficient which considers that not the whole mass of a roller is distributed over its circumference; it may be taken practically that $K = 0.8\text{-}0.9$; and g is the acceleration due to gravity, m/s^2.

Thus, the total resistance to the motion of a load possessing a gravity force G on a roller conveyer is the sum of the following components:

1. The resistance to load rolling on the rollers

$$W_1 = G2k/D \qquad (14.1)$$

where k is the coefficient of rolling friction of the load on rollers.

2. The resistance due to friction in roller axles; if the load lies on z' rollers, we have

$$W_2 = (G + Pz')\,\mu d/D \qquad (14.2)$$

where μ is the coefficient of friction in roller axles and d is the axle diameter.

3. The resistance due to sliding of the load on rollers (it accounts for the kinetic energy imparted to rollers by the load). For a single roller, the respective work is $2A$ and for all z rollers on a conveyer, the work is equal to $2Az$; therefore, this component of resistance on a conveyer of length L (m) (the path of load motion) will be

$$W_3 = 2Az/L = KPzv^2/(gL) \qquad (14.3)$$

The total force of resistance to the motion of a load, or the force required to move a load on an unpowered roller conveyer, is found as the sum of these components

$$\begin{aligned} W &= W_1 + W_2 + W_3 \\ &= 2Gk/D + (G + Pz')\,\mu d/D \\ &\quad + KPzv^2/(gL) \end{aligned} \qquad (14.4)$$

Hence we can find the coefficient of resistance to the motion of a load, w', or the tangent of angle β of a gravity-type roller conveyer on which the load will move at a constant speed v

$$\begin{aligned} w' &= \tan\beta = \frac{W}{G} \\ &= \frac{2k}{D} + \left(1 + \frac{Pz'}{G}\right)\frac{\mu d}{D} + \frac{KPzv^2}{gLG} \end{aligned} \qquad (14.5)$$

If, as shown in the diagram of Fig. 14.6*b*, the time interval between adjacent moving loads t_1 is shorter than the period of rotation of a roller under load t_1'' ($t_1 < t_1''$), the roller will not stop and the kinetic energy imparted to it is lower than in the previous case, so that the force of resistance and the required angle of decline of the conveyer will be smaller. If the time interval t_1 is equal to the period of uniform rotation of rollers FH, the force of resistance and the required angle of decline have the minimum values

$$\begin{aligned} w'_{\min} &= \tan\beta_{\min} = \frac{2k}{D} + \left(1 + \frac{Pz'}{G}\right)\frac{\mu d}{D} \\ &= \frac{2k + \mu d}{D} + \frac{Pz'}{G}\,\frac{\mu d}{D} \end{aligned} \qquad (14.5')$$

Irrespective of the method of delivery of unit loads onto a roller conveyer, its angle of decline can be smaller at a lower gravity force of the rotating parts of rollers and a greater mass of load, i.e. at a lower P/G ratio. For that reason, unpowered roller conveyers should be declined at a greater angle if a change is made from heavier to lighter load units. Of large importance is also the condition of the bottom surface of load units, which determines the coefficient of rolling friction k, and the type and state of bearings and the diameter D of rollers. Average slopes of unpowered roller conveyers (based on practical data) are given in Table 14.1. In curved conveyer sections, the slope should be increased by 0.5-1 %.

Table 14.1. Recommended Slopes of Unpowered Roller Conveyers

Kind of load	Mass of load units, kg	Slope, %
Steel sheets:		
straightened	30	1.0-1.5
unstraightened	30	2-3
riffled	30	2
Smooth treated castings	30	0.5-1.5
Iron pigs	30	3.5-4.0
Prepared moulds	10-50	1.5-2.5
Wooden bars	100-400	0.5-1.5
Boards:		
rough (sawn)	30	3-4
	15	4
planed	15	2.5
Sheet-metal containers	10-30	2-3
	30-150	2.0-2.5
	150-500	1.5-2.0
	500-1000	1.0-1.5
Planed-board boxes, wooden bottom-plates	10-25	2.0-2.5
	25-125	1.5-2.0
	500-1200	0-5-1.5
Cartons	1.5-3.0	6-7
	3-8	5-6
	8-25	4-5

14.3. Powered Roller Conveyers

14.3.1. Types and General Design

In powered roller conveyers, rollers are driven from a motor. Powered conveyers are widely used in rolling mill shops for transportation of metal to rolling mills, shears, saws, straighteners and coolers and to the store. They have found wide application in plants where metal sheets and sections are cut, in saw mills and wood recovery plants for conveying of logs and boards between machine tools, etc.

Roller conveyers in rolling mill shops can be divided into the working and transporting conveyers. The former are essentially the components of a rolling mill (infeed and outfeed conveyers) and the latter serve to transport metal in the shop.

By the method of driving the conveyer rollers, it may be distinguished between conveyers with a group drive and those with individually driven rollers. In a group drive, the driving force is transmitted to conveyer rollers by a longitudinal shaft, driving chains, a V-shaped or flat belt (Fig. 14.7). The scheme of a conveyer drive with a longitudinal shaft and angular gears arranged at the ends or roller axles is illustrated in Fig. 14.7*a*. The roller axles rotate in external bearings. Conveyers of this type are used in heavy-duty applications and are often made reversible (with a reversible motor or reversible transmission mechanism).

Figure 14.7*b* shows the scheme of conveyer drive by means of short driving chains; each chain connects two rollers and runs around sprockets mounted on the ends of the rollers.

Figure 14.7*c* shows a roller driven from a single chain running along the whole length of the conveyer (or a long conveyer section); the chain is prevented from running off the sprockets by a shield arranged above it. The return run of the chain slides in a trough. Various schemes of chain transmission emplo-

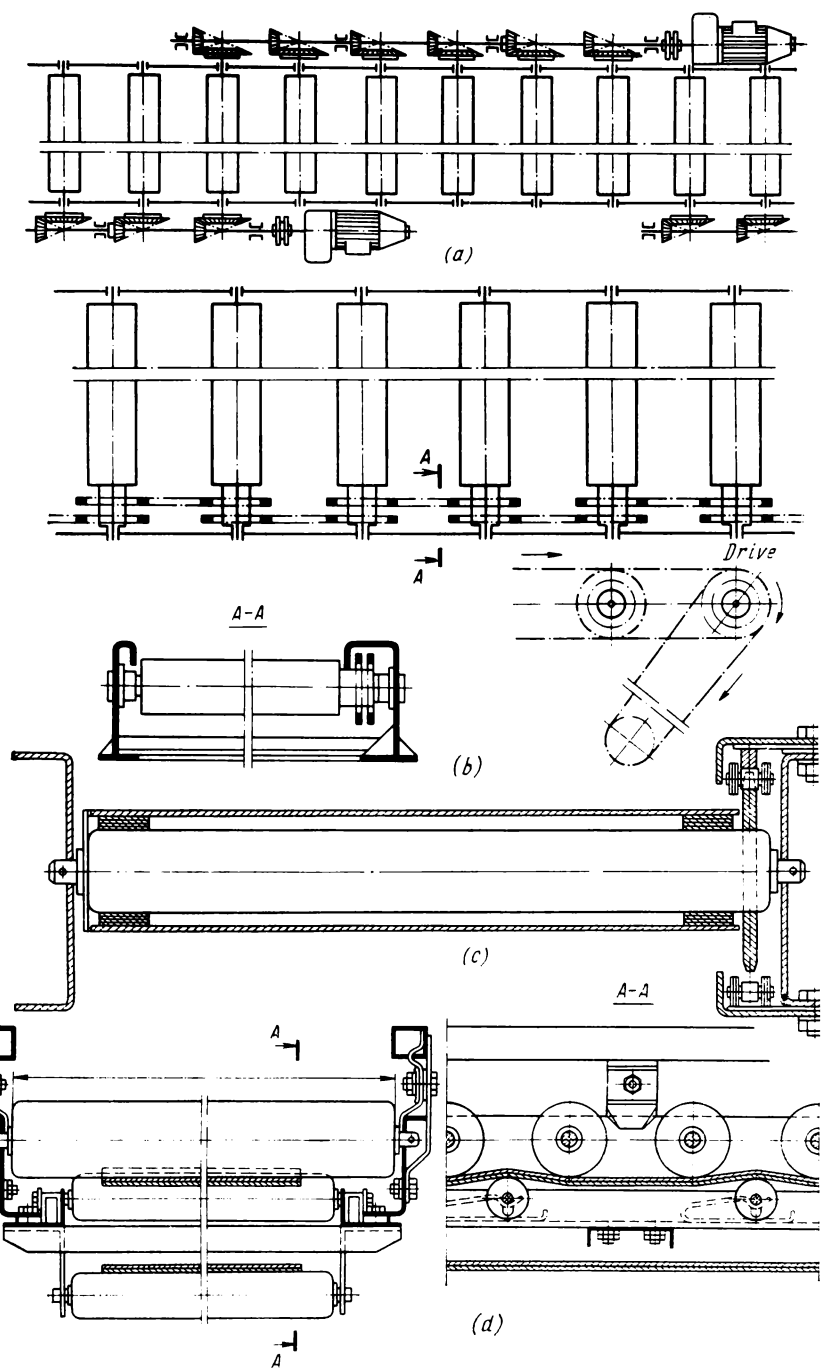


Fig. 14.7. Roller conveyer drives
(*a*) with longitudinal shaft and angular gear transmissions; (*b*) with 'roller-to-roller' drive chains a single drive chain; (*d*) belt drive

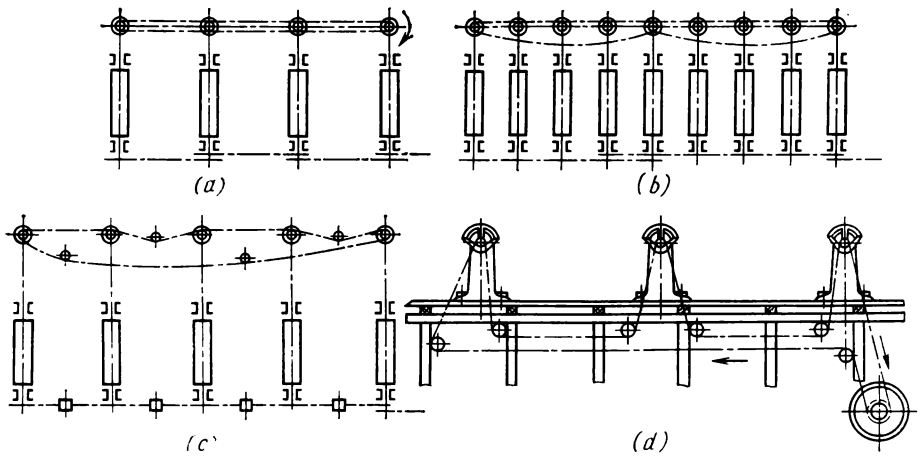


Fig. 14.8. Chain transmissions of powered roller conveyers
(*a*) 'roller-to-roller' short transmissions; (*b*) sectional chains for a group of rollers; (*c*) sectional chains with pressure rollers to increase the wrapping angle; (*d*) with chain wrapping the sprockets at an angle of 180°

yed, as a rule, under light operating conditions and low speeds of loads, are illustrated in Fig. 14.8.

A belt drive for roller conveyers is shown in Fig. 14.7*d*. The belt is arranged under the driven rollers and is pressed to them by auxiliary (snubbing) rollers, which also support the return run of the belt. Conveyers of this type are characterized by smooth and noiseless operation and can be employed efficiently under light and moderate operating conditions when a high speed of loads is essential. In some cases,

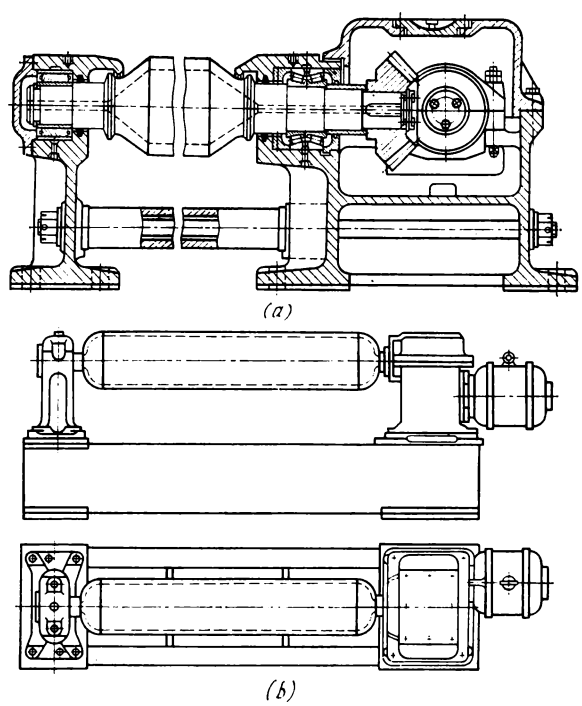


Fig. 14.9. Rollers of a metallurgical conveyer
(*a*) with group drive; (*b*) with individual drives

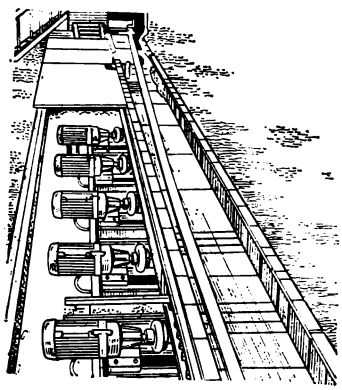


Fig. 14.10. Roller conveyer with individually driven rollers

unit loads on a roller conveyer should be turned during transportation. This is done by means of tapered rollers which impart a higher linear speed to loads at one side and a lower, at the opposite side. The length of the turning section in such conveyers is however substantial. Load units can be turned on their vertical axis on a short length of conveyer or even at a point by means of twist rollers: two shorter rollers are set on a common shaft and rotated by a chain or gear transmission in opposite directions.

Two types of conveyer rollers for heavy-duty operation (in rolling mill shops) are shown in Fig. 14.9. A conveyer of this type, with the floor removed, is shown in Fig. 14.10.

14.3.2. Calculation of Powered Roller Conveyers

Transporting conveyer. When a transporting conveyer is started, all rollers are rotated by the drive motor continuously irrespective of whether they carry loads or not. For a rated throughput capacity of a conveyer Q (t/h) (considering uneven delivery of loads), conveyer length L (m), its horizontal projection L_h (m), lifting height H (m), the number of rollers z, the gravity force of the rotating part of a roller P (N), and the speed of loads v (m/s), the required power of the drive motor (kW)

$$N = \left(\frac{QH}{360} + \frac{QL_h w'}{360} + \frac{zPw_1' v}{1000}\right)\frac{1}{\eta} \quad (14.6)$$

where w' is the coefficient of resistance to the motion of loads on the conveyer [see equation (2.23)]; w_1' is the coefficient of resistance to rotation of rollers which is obtained from equation (2.23) at $k = 0$ ($w_1' = \mu d/D$); and η is the efficiency of the drive mechanism.

For horizontal conveyers, $H = 0$ and $L_h = L$, and therefore, the drive power in kW

$$N = \left(\frac{QLw'}{360} + \frac{zPw_1' v}{1000}\right)\frac{1}{\eta} \quad (14.7)$$

For a given gravity force of loads, G (N), and their number per hour, Z [see equation (2.15)], the previous equation takes the form

$$N = \left(\frac{GZLw'}{3600 \times 1000} + \frac{zPw_1' v}{1000}\right)\frac{1}{\eta} \quad (14.7')$$

The power of the drive motor should be increased with an increase of the throughput capacity Z or the number of rollers z. The drive efficiency is usually taken equal to 0.8-0.85.

The time interval (s) between moving loads with their uniform delivery to a conveyer

$$t = 3600/Z \quad (14.8)$$

and the time (s) of their motion along the conveyer

$$T = L/v \quad (14.9)$$

The number of load units present at a time on the conveyer can be found by the formula

$$Z_0 = T/t = ZL/(3600v) \quad (14.10)$$

The required drive motor power

$$N = \frac{(Z_0 Gw' + zPw_1')v}{1000\eta} \quad (14.11)$$

If the number of load units charged onto a conveyer is low, it may turn out that $Z_0 < 1$, i.e. the conveyer pe-

riodically runs empty. In that case it should be taken in equation (14.11) that $Z_0 = 1$.

The highest torque (N m) which is transferred to a roller when a load unit lies simultaneously on i rollers and its gravity force is distributed between the rollers with a coefficient of nonuniformity $K_1 > 1$, can be determined by the formula:

$$M' = \left(\frac{K_1 G w'}{i} + P w_1'\right) \frac{D}{2} \quad (14.12)$$

If a load unit is retained by a stop and the rollers continue to rotate, the torque (N m)

$$M_1 = \left[\frac{K_1 G f}{i} + \left(\frac{K_1 G}{i} + P\right) w_1'\right] \frac{D}{2}$$

where f is the coefficient of friction of the load on roller surface.

In that case, since the force of sliding friction of rollers increases and the force of rolling friction decreases, the drive power (kW) of the conveyer, as found by equation (14.7), must be increased by

$$G\,(f - 2k/D)v/(1000\eta) \approx Gfv/(1000\eta) \quad (14.13)$$

Working conveyers. For working (infeed and outfeed) conveyers of rolling mills, which are characterized by frequent and quick changes of the direction of motion, it is essential to consider the additional forces acting during the periods of accelerated motion. It may be taken that the rotational frequency of conveyer rollers during such periods increases by the linear law, and therefore, the angular and linear accelerations at their circumference are constant. Two different cases will then be possible, depending on the acceleration which is determined by the power and starting characteristic of the motor and the force of resistance:

1. The force of friction with which a load unit of mass m is entrained by rollers into motion is equal to or greater than the inertia force of the load, i.e.

$$mgf_0 \geqslant mj_0 \text{ or } j_0 \leqslant gf_0 \quad (14.14)$$

where f_0 is the coefficient of friction of the load at rest relative to the surface of rollers, and j_0 is the linear acceleration of the load.

In that case the load will roll on the rollers without sliding, i.e. will move at a linear speed equal to the circumferential speed of the rollers.

2. The force of load friction on the rollers is smaller than the inertia force, so that the load rolls and slides on the rollers during the periods of acceleration. The acceleration of the load

$$j = gf \quad (14.15)$$

where f is the coefficient of sliding friction of the load on roller surface.

In that case, the load in the acceleration periods moves relative to rollers, since its mass has no effect on the dynamic forces acting on the conveyer but a constant friction force Gf acts between the load and rollers.

The angular acceleration of the rollers for the first and the second case is respectively:

$$(1)\varepsilon_0 = 2j_0/D \leqslant 2f_0 g/D \quad (14.16)$$

$$(2)\ \varepsilon = 2j/D = 2fg/D \quad (14.16')$$

With individually driven rollers (drive motors mounted on their axles), the maximum torque on the motor shaft [see equation (14.12)] is found as follows:

$$(1)\ M = M_{st} + M_d = M' + \left(J_r + J_m + \frac{K'G}{ig}\,\frac{D^2}{4}\right)\varepsilon_0 \quad (14.17)$$

$$(2)\ M = M_{st} + M_d = M' + \frac{K'Gf}{i}\,\frac{D}{2} + (J_r + J_m)\,\varepsilon \quad (14.17')$$

where J_r and J_m are the moments of inertia of the roller and motor rotor; K' is a coefficient to consider the portion of the gravity force of the load that is transmitted to the roller; $K' = 1$, if the roller pitch is greater than the length of load unit and $K' < 1$ otherwise.

With a group drive of a conveyer, all z rollers are put into rotation simultaneously. The static moment of the roller axle during the acceleration periods

$$(1)\quad M_{st} = (Gw' + zPw_1')\frac{D}{2}$$

$$= \left(G\frac{2k+\mu d}{D} + zP\frac{\mu d}{D}\right)\frac{D}{2}$$

$$= (Gk + G + zP)\frac{\mu d}{2}$$

$$(2)\quad M_{st} = Gk + (G + zP)\frac{\mu d}{2} + Gf\frac{D}{2} \qquad (14.18')$$

where k is the coefficient of rolling friction.

The dynamic moment on the roller axle

$$(1)\quad M_d = \left(J_r z + \frac{GD^2}{4g}\right)\varepsilon_0 \qquad (14.19)$$

$$(2)\quad M_d = J_r z\varepsilon \qquad (14.19')$$

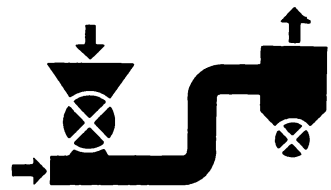
If K_{tr} is the transmission ratio from the motor shaft to the roller axle and η is the efficiency of the transmission mechanism, the total torque of the motor during the periods of acceleration

$$M_m = (M_{st} + M_d)\frac{1}{Kt_r\mu} + J_m\varepsilon_m C \qquad (14.20)$$

where ε_m is the angular acceleration of the motor shaft, which is greater by a factor K_{tr} than the acceleration of roller axles, and C is a coefficient which considers the moment of inertia of rotating masses of the transmission mechanism (approximately $C = 1.1\text{-}1.2$).

During acceleration periods, the drive motor develops the highest power at the end of a period when its rotational frequency reaches the upper limit. When selecting the power of a drive motor, it is essential to consider its overload capacity. The starting torque of motors is usually 1.8-2 times the rated torque in steady operation.

15
Hydraulic and Pneumatic Transport Systems



15.1. Hydraulic Transport Systems

15.1.1. Design

In hydraulic transport installations (which may be called 'hydraulic conveyers'), bulk loads are carried in a pipe or trough by the stream of a liquid (usually water). The mixture of a material with water is called hydraulic mixture, or pulp. The thickness and fluidity of pulp is determined by the ratio between the solid and the liquid component in it.

In troughs (cannels), the pulp is moved by gravity and the trough must be declined in the direction of pulp motion.

In pipes, the pulp can be carried by gravity or due to the pressure of the liquid created by a pump. In the latter case, the material can also be transported horizontally or upwards.

Hydraulic transport of bulk loads is used widely in the industry. In boiler plants, it is employed for the removal of slags and ash, in metallurgy—for removal of slags from furnaces, in mining —for transportation and lifting of coal, ores and other minerals and for delivery of filling materials to stow worked-out areas, and in civil engineering—for conveying of soil washed off by water jets.

Hydraulic transport offers certain substantial advantages. In particular, hydraulic transport systems have a high throughput capacity, can carry loads to appreciable lengths without overloading on intricate paths inclined at any angle or vertical; their mechanical equipment is not numerous and is concentrated at a few points of the path, and therefore, is easy for maintenance; transportation can be combined with some technological processes ('wet' preparation of ores, quenching and pelletizing of slags, screening, etc.); the transportation process can be completely controlled automatically to minimize hand labour. For these reasons, hydraulic transport systems in many cases turn out to be more efficient economically than mechanical transport means, such as conveyers, automobiles or railway cars.

The limiting factors for the application of hydraulic transport systems are the kind and characteristics of the loads, in particular, their particle size (in some cases, preliminary crushing can be needed); high wear of pipelines and mechanical elements which come in contact with the pulp, especially in transport of abrasive loads; an elevated power consumption; a high consumption of water; problems of water freezing in winter time; adverse effect on humidity in closed rooms.

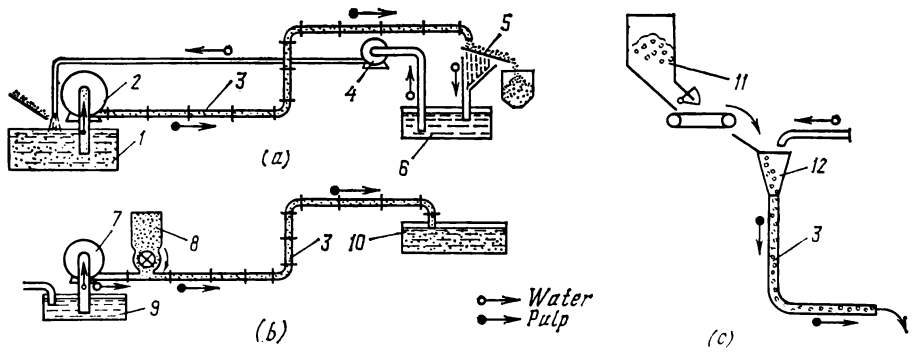


Fig. 15.1. Schemes of hydraulic transport systems
(*a*) with pulp pump; (*b*) with water pump and feeder; (*c*) gravity type; *1*—water conduit; *2*—pulp pump; *3*—pulp pipeline; *4*—clarified water pump; *5*—dewatering screen; *6*—pulp tank; *7*—water pump; *8*—gangue bin with feeder; *9*—water tank; *10*—pulp tank; *11*—gangue bin; *12*—mixing funnel

15.1.2. Schemes of Hydraulic Transport Systems

Hydraulic pressure transport systems can be classified most conveniently by the method of feeding of the material to be transported into the transport pipeline, which determines the mechanical equipment needed for the purpose. The principal schemes of systems are illustrated in Fig. 15.1*a* and *b*. The system in Fig. 15.1*a* has a pulp pump which pumps the pulp from a tank through the pipeline. In the scheme shown in Fig. 15.1*b*, a water pump takes water from a water tank and pumps it through the pipeline, whereas the material to be transported is fed into the pipeline by a separate feeder, in the case considered, by a hermetically sealed rotary drum feeder. An advantage of the first scheme is that it requires no intricate feeder. But the main mechanical unit (pump) in the second scheme operates on pure water, and therefore, the pump may be of a simpler design and, what is more important, is less subject to wear and damage by solid particles.

At the final destination point of a system, the pulp can be discharged from the pipeline directly into a receiving tank (as shown in Fig. 15.1*b*) or onto a screen to separate the material from water. The clarified water from the receiving tank can be repumped, if required, into the pulp tank and then fed into the pipeline, i.e. it moves in a closed cycle.

In some cases, when the required pressure can be formed due to a difference of levels (for instance, when a filler material should be lowered in a mine and distributed underground to stow the worked-out areas), the bulk material and water can be delivered together into a receiving mixing funnel (Fig. 15.1*c*).

The scheme of a hydraulic transport system for removal of slag and ash from a boiler plant is shown in Fig. 15.2*a*. A water pump *5* pumps water under a high pressure into a water pipeline *6* which has branched pipelines to slag and ash collectors *1*. Ash and slag are washed off from the collectors by means of ejectors and moved through a collecting pulpline *2* to a dump or, as shown in Fig. 15.2*a*, into settling tanks *4* from which they are loaded into railway cars by a grab-bucket crane *3*.

An ejector (a jet device) of the hydraulic system is illustrated in Fig. 15.2*b*. A water jet under pressure is ejected at a high speed from a conical nozzle and entrains ash or slag from a feeding funnel due to a vacuum formed around it. Water and material are intermixed to form pulp in a short pipe (diffuser) arranged at the nozzle, where the kinetic energy of the jet is again transformed

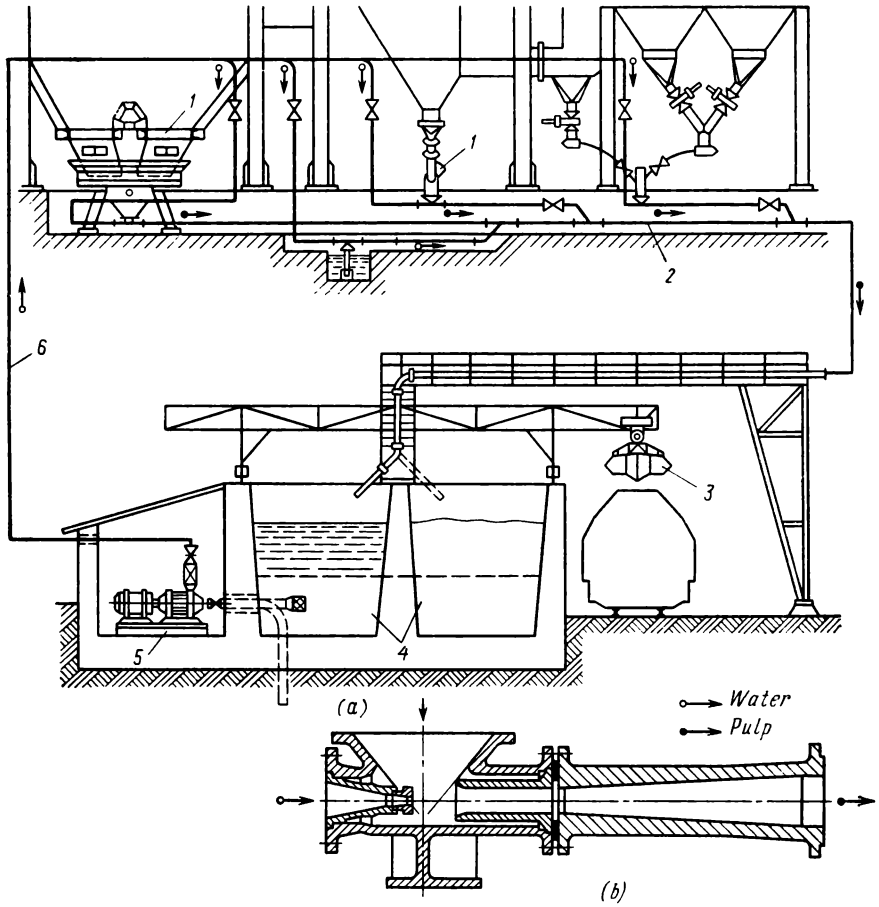


Fig. 15.2. Hydraulic transport system for slag and ash removal from a boiler plant (a) scheme; (b) ejector

into the potential energy of pressure which is needed for further transportation of the pulp through the pulpline.

Figure 15.3 shows an installation which supplies a filler material (slag, crushed gangue, etc.) to stow work-out faces in a mine. The installation operates by utilizing the natural head which forms in the vertical section of the pipeline arranged in the mine. It comprises a belt conveyer *1* to supply the filler material, a stationary gangue screen *3* onto which water is supplied from the top through a pipeline *2*, a roll crusher which disintegrates larger lumps retained by the screen, a receiving mixing funnel *10* mounted under the screen and a pipeline *9* arranged inside a water chamber. The pipeline *9* has longitudinal slits in the bottom portion through which air can pass out and water can flow in to control the pulp level in it. The pipelines are provided with water and pulp flowmeters (*5* and *7*); a pressure gauge *8* measures pressure in the pulpline.

15.1.3. Mechanical Equipment of Hydraulic Transport Systems

The mechanical equipment of hydraulic transport systems includes various feeders and pumps.

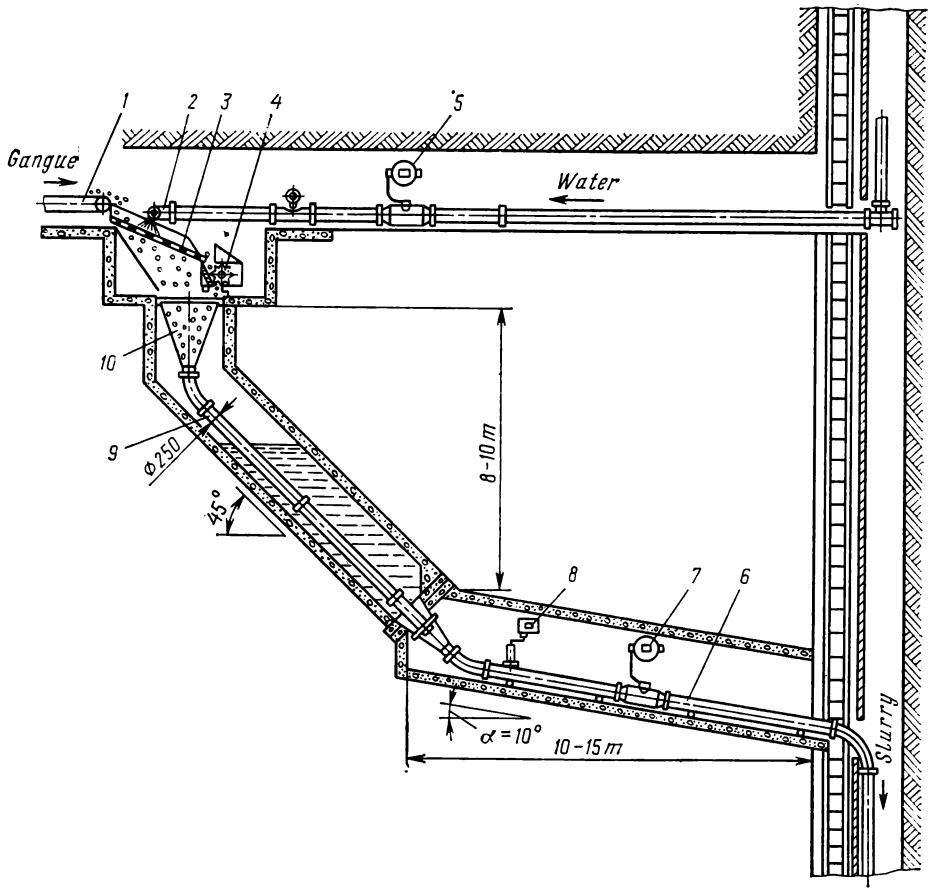


Fig. 15.3. Hydraulic transport installation for stowing work-out areas in mines

Feeders (loading devices) should introduce the material to be transported into the pipeline which is under a high pressure of water, and should not allow the backward motion of water. This is achieved by one of the two methods: (1) a feeder under an open bin or funnel pushes the material mechanically into the pipeline against the water pressure (continuous-type feeders) or (2) the material is passed through one or more chambers with gate valves at the top and bottom which are opened and closed intermittently to make the pressure in them equal to that in the pipeline (intermittent feeders, or box feeders). Box feeders operate cyclically, the whole cycle including the time for filling and emptying the chamber (or chambers) and for successively opening and closing their gate valves. For more uniform (sometimes almost continuous) supply of the material into the pipeline, box-type feeders are usually made consisting of two parallel sections, so that one of the sections can be filled with the material while the other discharges it into the pipeline.

A continuous screw-type feeder (Fig. 15.4) has three main units: drive *3* (motor, turbo-clutch and reducer gear), screw *2* in a cylindrical housing in which the screw rotates, and pipeline *1*. The housing is connected at one end to a receiving funnel and at the other end to the pipeline and forms a mixing

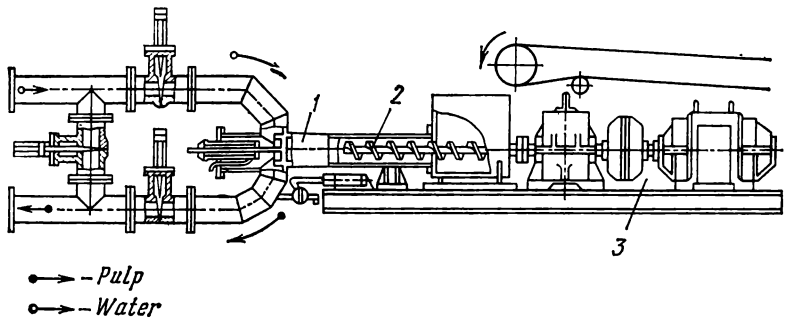


Fig. 15.4. Screw feeder for a hydraulic transport system

chamber. Gate valves in the pipeline can disconnect the latter from the feeder or let water flow by-passing the mixing chamber.

A bulk material (say, coal) is fed into the receiving funnel by a belt conveyer and then transferred by the screw in the cylindrical housing into the mixing chamber where the material is mixed with water and the mixture is conveyed through the pipeline. The screw compacts the material in the mixing chamber at the end of the cylindrical housing (for better effect, the screw is sometimes made with a decreasing lead), so that water cannot penetrate in the reverse direction into the funnel. Water penetration into the funnel can be prevented reliably if the rate of coal feeding by the screw exceeds the rate of water filtration through the layer of compacted coal. For that reason, screw feeders are not employed with materials consisting of hard unbreakable lumps, since water can penetrate quickly through spaces between them. Feeders of this type are also not recommended for abrasive loads which can cause quick wear of the screw and housing.

Screw feeders have a high unit power consumption, but are subject to wear. If the useful work is the work for introducing the material into the pipeline and displacing water from the feeder (which is proportional to the material volume per unit time multiplied by the water pressure in the pipeline), the efficiency of screw feeders in terms of the installed motor power does not exceed 20-30%.

The advantages of screw feeders are continuous action and small dimensions. In view of their drawbacks (a relatively low throughput capacity and certain problems of tightening), they are employed in stationary plants less widely than box-type feeders in which a bulk material can be 'sluiced' from an open bin into a high-pressure pipeline.

A box-type feeder consisting of two parallel sections with two chambers in each of them is shown in Fig. 15.5.

The upper ('sluicing') chamber in each section is provided with gate valves at the top and bottom. The lower chamber has no valve at the bottom, but communicates immediately with the pipeline. The bulk load (coal or ore with lump size up to 70 mm) falls from a belt conveyer into a two-sector tilting meter *1* in which the volume of each 'pocket' is equal to the useful volume of the sluicing chamber *3*. When the material is being fed from funnel *2* into this chamber, the top valve is open and the bottom valve is closed. Upon filling the upper chamber, the top valve is closed automatically and water under pressure is admitted into the upper chamber to equalize the pressure in both chambers, after which the bottom valve is opened and the material falls into the lower mixing chamber *4* and further into the pulp

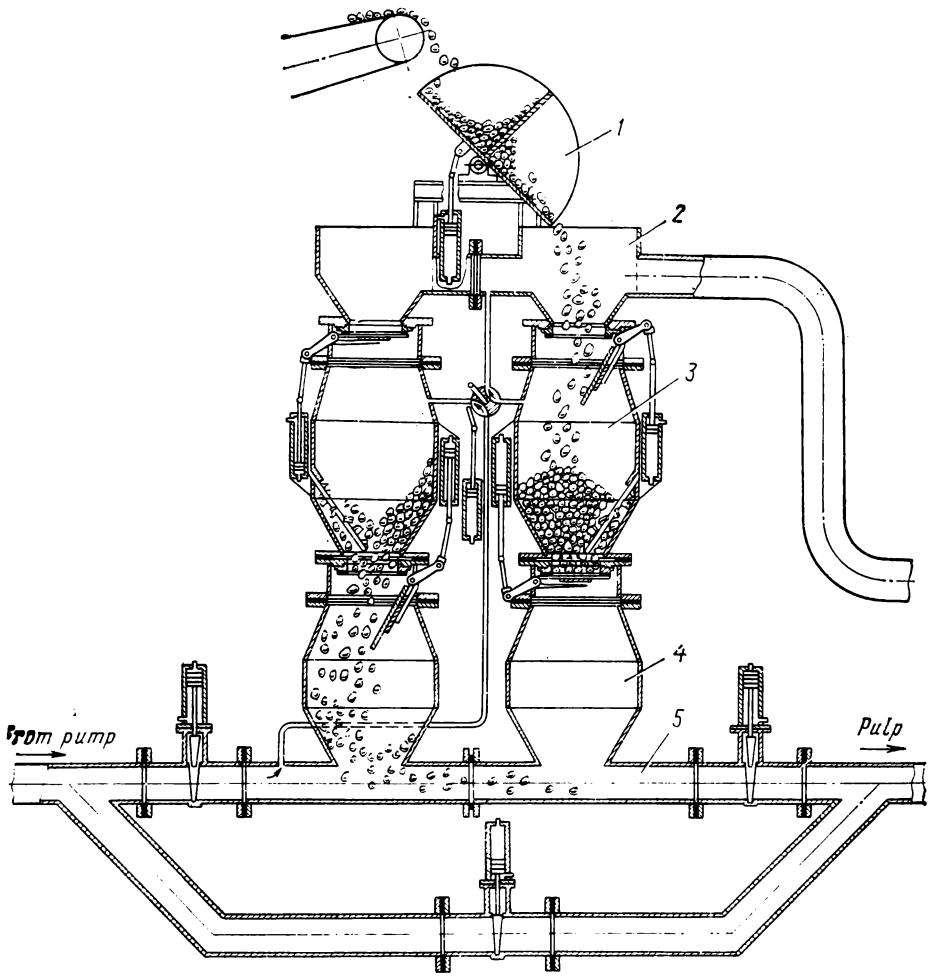


Fig. 15.5. Double-box feeder for a hydraulic transport system

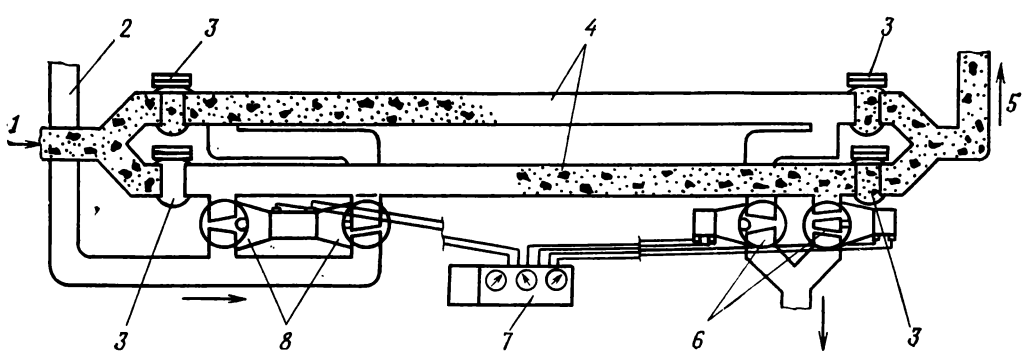


Fig. 15.6. Scheme of a box-tube feeder

As the first section discharges the material into the pipeline, the second section is filled with the material, i.e. their operation is shifted in time by half the cycle duration. The meters and gate valves are actuated automatically by hydraulic cylinders.

Now, tubular box-type (box-tube) feeders are employed increasingly in high-pressure hydraulic systems for transport of lumpy loads. They are simple in design, operate automatically and can handle materials with a relatively large size of lumps.

A box-tube feeder with two tubular chambers *4* and fittings is shown in Fig. 15.6. The tubular chambers communicate at one end with pipelines *1* and *2* to supply pulp and water and at the other end, with the main conveying pipeline *5*. The pulp is introduced into one of the chambers and then washed off by water, which is controlled by four automatic non-return valves *3*. The supply and removal of water are controlled by a program by means of gate valves *6* and *8* which are actuated from a panel *7*. Thus, the processes of successive filling of chambers with the pulp and discharging it into the main pipeline occur almost without interruption, i.e. practically continuously.

Pulp pumps are mostly of the centrifugal type; piston pumps are employed rarely and only with fine non-abrasive bulk materials. Piston pumps suffer from increased wear, have relatively large dimensions, and operate in a pulsating mode which can result in separation of solid fractions from the pulp. They are only advantageous in being capable of developing high pressures.

Centrifugal pulp pumps differ but little from water pumps in the design and operating principle. They may be of the single-, double- or multiple-stage type, depending on the number of impellers arranged in the pump housing. Single-stage pulp pumps are most common, and double-stage pumps are employed only rarely when a higher pressure should be developed. Double-stage pumps are more intricate in design and more subject to wear and produce a strong crushing effect on some kinds of loads, such as coal.

Pulp pumps should meet certain specific requirements, in particular, allow transportation of rather large lumps (practically up to 100 mm), have a high wear resistance, ensure easy repair and replacement of eroded parts, especially of sealings; the material should not be crushed excessively by the impeller and in further motion through a pump. In order to increase the service life of pulp pumps, the impellers and other critical parts are made of special steels or high-chrome cast iron, faced with hard alloys or lined with rubber.

Figure 15.7 shows the longitudinal section through a rubber-lined pump for handling especially aggressive hydraulic mixtures. The pump housing and shaft bearings are fastened on a common frame *7* with cover *1* which forms a central inlet pipe. A vertical pressure pipe *3* is in the periphery of the housing. An impeller *2* is set on the end of a shaft with a soft sealing with pipe connection *5* through which washing liquid can be supplied to clean the sealing. The housing, cover and impeller are lined with wear-resistant rubber *4*. The impeller shaft is rotated through a V-belt transmission *6* from an electric motor mounted near the pump or beneath it. Pumps of this type are designed for a relatively low flow rate of pulp (up to 300-400 m^3/h) and low pressures (up to 0.5 MPa). Suction dredge pumps can develop substantially higher flow rates and pressures: up to 4000 m^3/h and 0.66 MPa or in special designs, up to 7000 m^3/h and 0.71 MPa.

The working pressure in hydraulic transport systems is sometimes increased by connecting two single-stage pulp pumps in series. In systems of a large length or substantial lifting height, the pipeline can be divided into sections provided with intermediate transfer pumps. The sections of a pipeline can

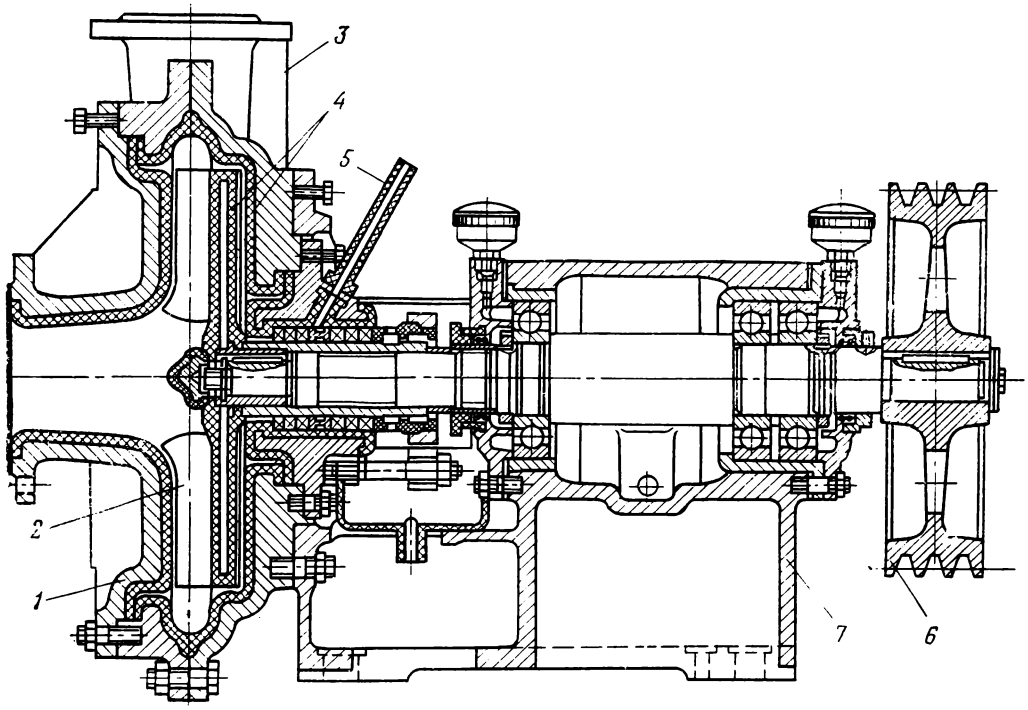


Fig. 15.7. Rubber-lined pump for aggressive hydraulic mixtures

be connected either directly, i.e. the end of a pipe section is connected immediately to inlet pipe of the pump of the next section, or through intermediate reservoirs (sumps) from which the pulp is sucked in by pumps. The first method involves lower hydraulic losses, but requires a more coordinate operation of all the pumps which are arranged at substantial distances from one another.

15.1.4. Automatic Control of Hydraulic Transport Systems

The main objective of automatic control in hydraulic transport systems is to maintain the best (most favourable) operating conditions. In the general case, the optimal conditions are obtained at a particular thickness (concentration) and speed of the pulp when the flow rate of the working liquid for transportation of a bulk load does not exceed the flow rate required to obtain the desired throughput capacity for the solid fraction and ensure steady and reliable operation of the system. Under the optimal conditions, the power consumption for pulp transportation and water recycling is, as a rule, lower.

The power consumption in a system depends on the pressure (head) losses per unit length of the pipeline. In horizontal pipelines, the head loss is only due to the resistance to the motion of the pulp.

As may be seen from the experimental curves shown in Fig. 15.8, head losses increase with increasing thickness of the pulp, though the total volume of the pumped pulp and water decreases substantially more quickly with an increase of pulp thickness, therefore the pulp with a rather high thickness is more suitable. As regards the effect of pulp speed, it may be seen from the diagram in Fig. 15.8 that the

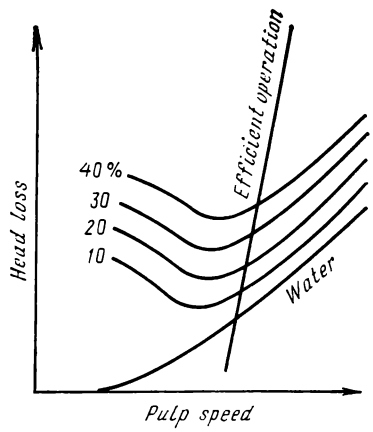


Fig. 15.8. Effect of concentration (%) and speed of pulp on head losses

head loss for water increases almost in direct proportion to its speed, while the head loss for pulp first decreases with increasing speed (up to a value at which solid particles are carried in the pulp in the suspended state) and then increases proportional to the speed. Thus, the conditions of the best economic efficiency are obtained at pulp speeds slightly higher than those at which the head loss is at the minimum, i.e. roughly along the intersecting line in Fig. 15.8.

The allowable concentration of a bulk material in the pulp depends mainly on the properties of that material. For light and easy-moving materials (such as peat or fine coal) it may be higher than for those which are less movable (say, ore). In some cases, the properties of a material may vary during transportation, thus involving variations in head losses which can require certain corrections in the pulp concentration along the pipeline.

Let us show how the pulp concentration can be varied automatically depending on the actual head loss, and therefore, on the actual resistance, in the horizontal pipeline of a system with a natural head in a vertical (or inclined) pipe section (see Fig. 15.1*c*).

The pressure at the pipeline exit is close to the atmospheric, and therefore, the velocity head of the stream at the beginning of the horizontal pipeline should be roughly equal to the resistance along its entire length. A remote-reading pressure gauge mounted in that place can serve as a sensor to measure the resistance and send a control pulse upstream of the pipeline to the loading station. The latter comprises a bin, a feeder under it, a mixing funnel, and a pipeline.

At a constant quantity of water fed into the system, the pulp concentration can be controlled automatically by varying the quantity of the material charged into the funnel by means of a gate valve or by changing the speed of the feeder (this can be effected most simply in shaking feeders). The feeder should diminish the quantity of charged material with an increase of the pipeline resistance above a certain limit, and vice versa.

Another problem of control in hydraulic transport systems consists in establishing the normal operating conditions as the flow rate of the solid fraction and pulp varies for some or other reason (as for instance, in the removal of slag pulp from boiler plants). In systems with a pulp pump, this can be effected most simply by controlling the rotational speed of the pump impeller. Two such systems in which the throughput capacity is controlled automatically depending on the level of pulp in reservoir *1* (from which the pulp is sucked in by a pulp pump) are shown schematically in Fig. 15.9. In the system in Fig. 15.9*a*, the rotational speed of the impeller of pulp pump *8* is varied by a hydraulic coupling *3* between the impeller shaft and squirrel-cage asynchronous motor *4*; in the system in Fig. 15.9*b*, this is done by varying the rotational frequency of a phase-rotor motor *4*. In either case, a control pulse is sent by a pulp level gauge *1* in reservoir *2* to a regulator *7*. In the system in Fig. 15.9*a*, the regulator sends a pulse to an actuating mechanism with a servomotor *6* which acts

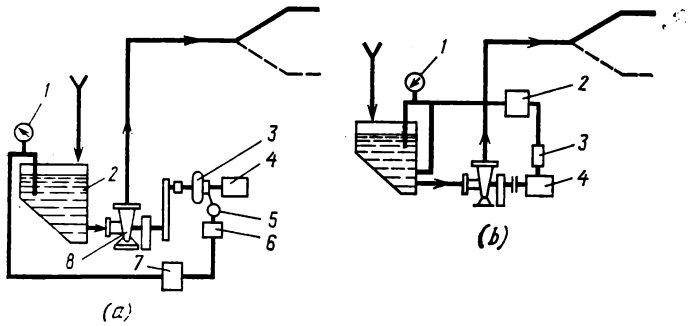


Fig. 15.9. Schemes of hydraulic transport systems with pulp pump and automatic control of throughput capacity
(*a*) with hydraulic coupling; (*b*) with speed-controlled electric motor

on the lever of speed regulator *5* on the driven shaft of hydraulic coupling *3*.

In the system in Fig. 15.9*b*, a regulator *2* sends a pulse to a drum controller *3* which varies the rotational frequency of phase-rotor motor *4*.

For smooth variation of the speed of pulp pump impeller in a wide range, it is possible to employ a positive-displacement hydraulic transmission possessing a high efficiency. Torque converters and combinations of a rectifier and d.c. motor can be employed for the purpose.

In some plants, the desired thickness of the pulp is controlled by varying the proportion of the solid and liquid components. In systems with box-type feeders, it is possible to control the degree of filling and the cycle time of the feeder or the quantity of water pumped into the pipeline. In both cases, special instruments for automatic measurement of pulp thickness (consistometers) are used as sensors in the control system. Among them, radiometric sensors based on the exposure of the pipeline to gamma-rays, are most promising. As gamma-rays pass through a pulpline, they are attenuated depending on the concentration of the solid component in the pulp. The penetrated gamma-rays induce an ionization current in an ionization chamber at the opposite side of pipeline, which is proportional to the intensity of penetrated gamma-rays. The current is amplified and fed into a measuring instrument whose scale is calibrated in per cent units of the concentration of solid component in the hydraulic mixture. The feeding devices of the hydraulic transport system can receive control pulses from this instrument.

15.2. Pneumatic Transport Systems

15.2.1. Design and Characteristics

Pneumatic transport systems are much similar to the hydraulic type and differ only in that the material is carried in a stream of air, rather than of water. Air in the pipeline moves due to a pressure difference between the beginning and end of the pipeline, which is produced by a pressure blower or vacuum fan. Air moves at a high speed through the pipeline and forms with a powdered or fine-size lumpy material a more or less homogeneous air mixture. Material lumps and particles usually move jumpwise in the suspended state and partially slide along the pipe bottom.

A pressure difference in pressure pipelines is formed by compressors, air blowers and fans and in vacuum systems, by vacuum pumps and (exhaust) fans.

Pneumatic transport finds wide application in various branches of industry, civil engineering, agriculture, and materials-handling operations in railway and water transport. Among the most common bulk loads transported pneumatically are cement, pulverized and fine- and medium-size coal, blast-

furnace dust, ground ore, grain, apatite concentrate, salt, milled peat, ash and slags, cotton, sawdust and chips, phosphate rock and many other pulverized and fine-size lumpy materials.

The throughput capacity of pneumatic systems may be up to 100 t/h and rarely, above that value and the transportation length, tens or hundreds metres and in some cases, up to 2 km or more.

Pneumatic transport systems are employed widely due to their numerous advantages which are essentially the same as those of hydraulic system, in particular; they are perfectly tight; loads are transported without losses, and therefore, are not influenced by the surroundings; transportation paths may be of any shape, with easy connection of horizontal, vertical and inclined sections; mechanical equipment of systems is concentrated at a single point and can be maintained easily; transport lines can be branched to deliver loads from a few points to a single destination or from a single point to a number of destinations; transportation can be combined with some technological processes, such as drying, cooling, etc. Pneumatic transport ensures hygienic and safe conditions for the personnel when conveying harmful chemical products and hot or odorous materials.

The main drawbacks of pneumatic transport include a high unit power consumption and intensive wear of pipes (especially in turns) and other elements in contact with the flow of material. Efficient application of pneumatic transport is limited not only by the size of lumps (usually 30-40 mm and sometimes up to 80 mm), but also by some properties of the transported materials, such as moisture content or stickiness (their ability to adhere to the walls of pipes, tanks, feeders, etc.). Powdered and especially fine-dispersed materials with particle size less than 10 μm are difficult to separate from air at the destination points.

Apart from the above pneumatic transport systems for carrying bulk materials, there are some other types, for instance, the pneumatic tube (mail) system for transport of small articles or small quantities of bulk materials in special cylindrical containers which close the whole cross section of the pipeline. In contrast to the main method of pneumatic transport in an air stream, containers in pneumatic mail systems are moved by the static pressure of air which acts on their end faces.

Another variety of pneumatic transport is represented by air-activated chutes and conveyers. The bottom of a chute is perforated, so that when air is blown from beneath, it activates (aerates) a bulk material in the chute, which thus acquires greater mobility and can move by gravity even in a slightly inclined chute. This principle is also used in air-activated pneumatic transport pipelines (see later in the chapter).

15.2.2. Schemes of Pneumatic Transport Systems

By the method in which a pressure difference is formed in a pipeline, all pneumatic transport systems can be divided into three substantially different groups: negative-pressure (vacuum) systems, positive-pressure (or simply pressure) systems, and a combined type (Fig. 15.10). In vacuum systems, the materials are moved due to air rarefaction; in pressure systems, they are carried in a stream of compressed air; in combined systems, the negative-pressure principle is used in some portions of the conveying path and the positive-pressure principle, in other portions.

In vacuum systems, the material is usually introduced into the pipeline through a suction nozzle and separated at the destination point by the 'sluicing' principle or in another way. In pressure systems, the material is fed by means of feeders operating by the 'sluicing' principle.

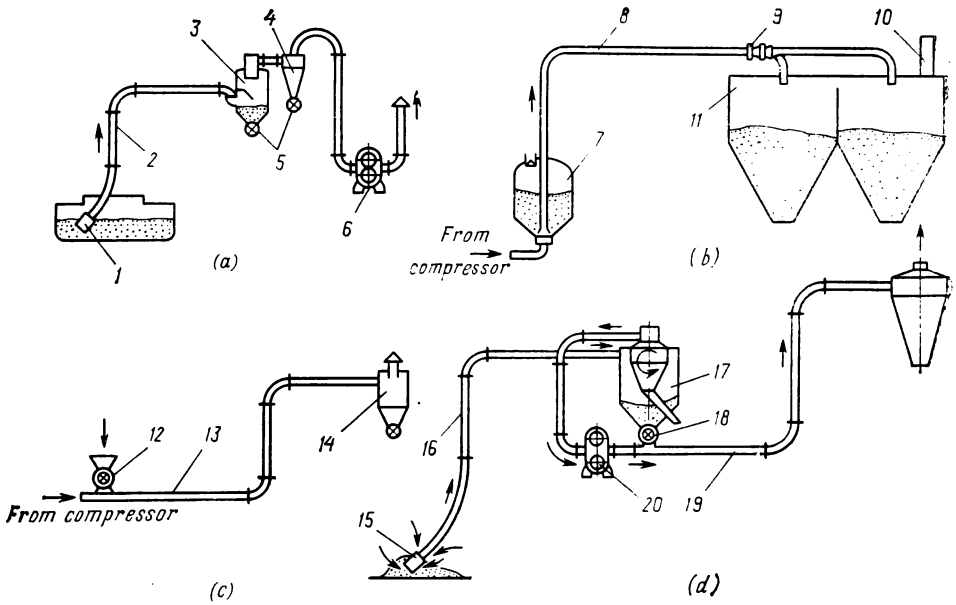


Fig. 15.10. Schemes of pneumatic transport systems
(*a*) vacuum-type; (*b*) and (*c*) pressure-type; (*d*) combined

In pipelines of pressure systems, the carrying air is compressed, i.e. is at a pressure higher than the atmospheric, so that the pressure difference and the air stream velocity can be higher than in vacuum systems, and therefore, the 'carrying capacity' of the air is higher than in vacuum systems. For that reason, pressure systems are more efficient for conveying poorly movable materials or when it is needed to carry the materials to a large distance or height. On the other hand, vacuum systems are employed preferably with easily moving materials (such as grain) and require no feeders.

In vacuum pneumatic systems, the mechanical components (air blower and separators) are concentrated at the end of the pipeline. In contrast to this, the blower and mechanical components (feeders) in pressure systems are arranged at the beginning of the pipeline. In combined-type systems, the blower and mechanical components are mounted somewhere in the mid of the conveying path.

As has been given earlier, an essential advantage of pneumatic transport systems is that the conveying path can be branched easily, i.e. it is possible to take the material from a number of points or to carry it to a number of destination points. In that respect, vacuum pneumatic systems can be branched more simply at the starting end and pressure systems, at the destination end, while combined-type systems can be branched at both ends.

In a vacuum pneumatic system (Fig. 15.10*a*), the material (grain from a barge in the case illustrated) is taken off by a suction pipe (nozzle) *1* and conveyed through pipeline *2* to a receiving separator *3* where the material separates from the air flow and falls to the separator bottom, since the air stream loses its speed. The air is then sucked off into another vessel of a smaller volume—a dust-collecting cyclone *4*, passes through an air blower *6* and is ejected into the atmosphere through a discharge pipe, while the separated material is discharged from the separator and dust

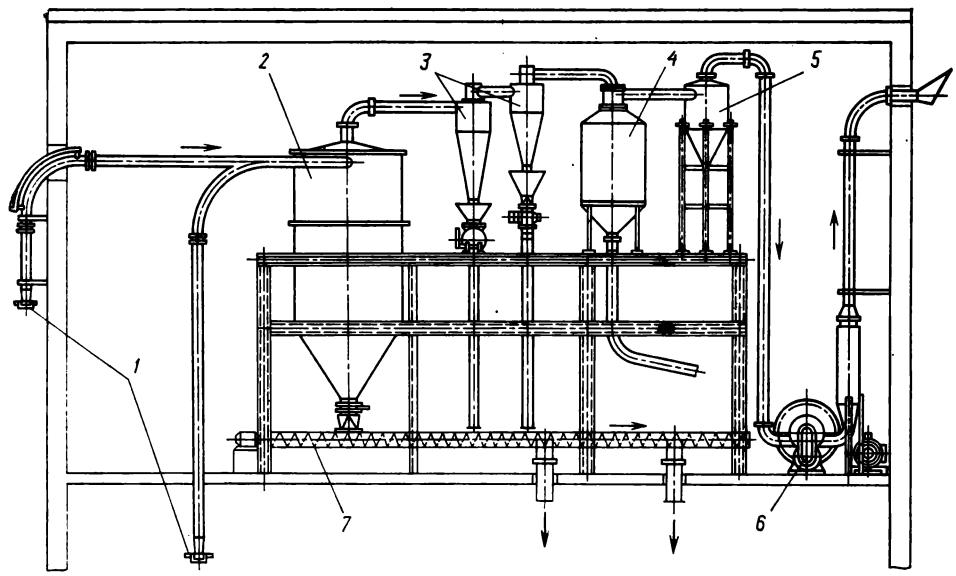


Fig. 15.11. Scheme of a vacuum-type pneumatic transport system for pulverized loads

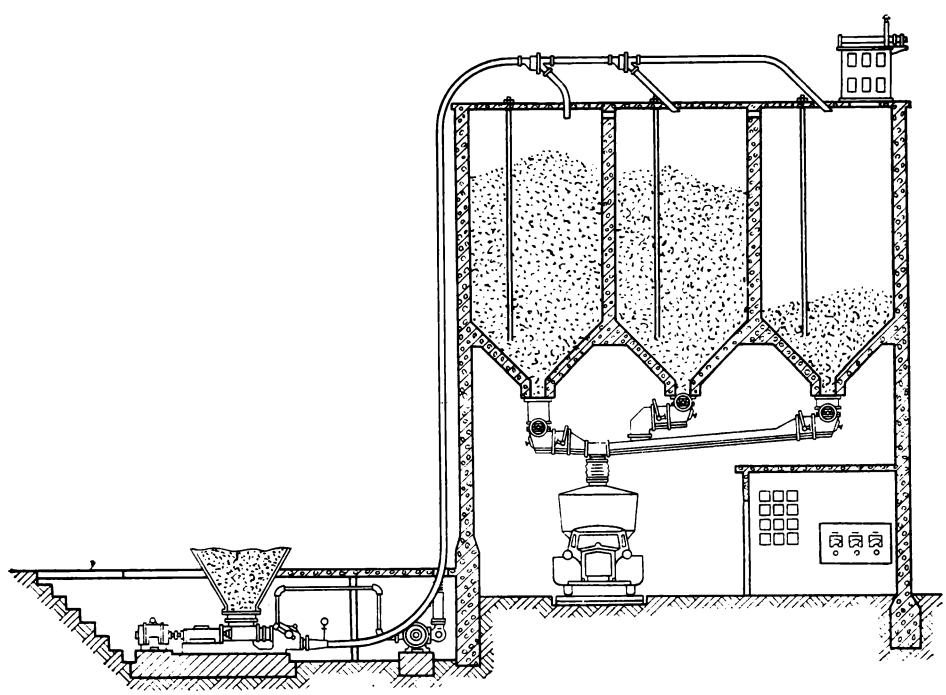


Fig. 15.12. Pressure-type pneumatic system with storage bins

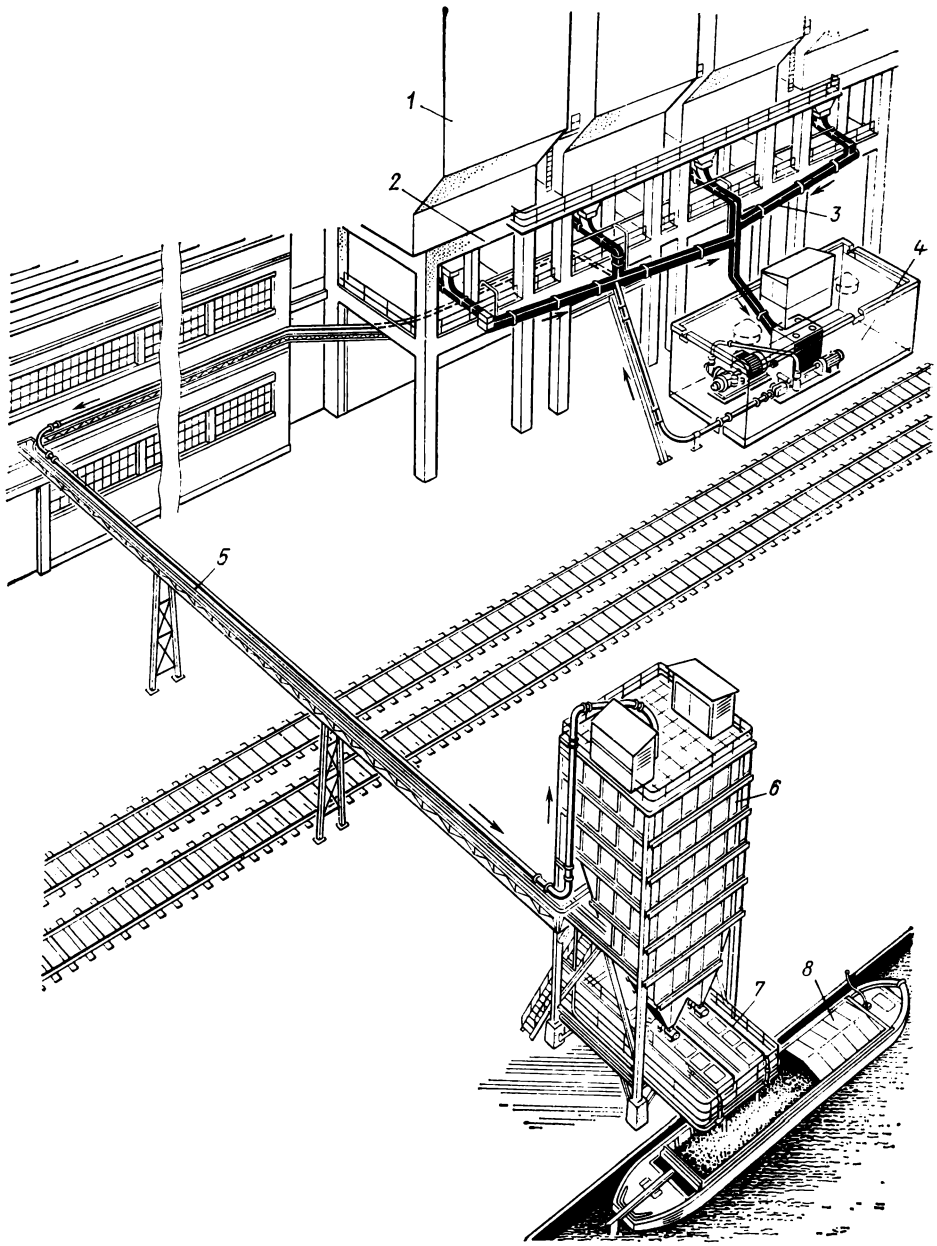


Fig. 15.13. Combined vacuum-pressure pneumatic transport system for ash removal from boiler plant

collector through sluicing valves *5* (in the form of rotaiing drum with cells). Vacuum systems of this type usually have two-four suction pipes with flexible hoses and nozzles.

In high-pressure pneumatic systems (for a pressure of 0.3-0.4 MPa or more), compressed air from a compressor moves through a pipeline into a closed reservoir—box-type feeder *7* (Fig. 15.10*b*) where the material is caught by the air stream, carried through pipeline *8*, and directed by a switch *9* into one of bunkers *11*. The bunkers serve as separators and are provided with filters *10* through which the air is discharged into the atmosphere.

In medium-pressure pneumatic systems (for a pressure of 0.2-0.3 MPa), the material is introduced into a compressed-air pipeline by a drum-type feeder *12* (Fig. 15.10*c*) and carried through pipeline *13* to a separator *14*. As in the previous case, the pipeline may have branches at the discharge end.

In a combined-vacuum-pressure system (Fig. 15.10*d*), the material is sucked off through a nozzle *15* into a pipeline *16*, enters a separator *17* at an intermediate point of the path, and is transferred through a sluicing valve *18* into a pressure pipeline. The blower *20* at the intermediate point sucks in the air from pipeline *16* and discharges it into pipeline *19*. The system can be branched at both ends.

Figure 15.11 shows the mechanical equipment of a vacuum-type pneumatic transport system for conveying pulverized and dusty materials, in which the air is cleaned off thoroughly before being discharged into the atmosphere. The air mixture is transported through pipelines *1* to a separator *2* and then passes successively through two dust-collecting cyclones *3*, scrubber *4* and moisture separator *5*. The cleaned air passes through a blower *6* and is discharged into the atmosphere. The material collected in the separator and two cyclones passes through sluicing valves and is fed into a screw conveyer *7* to be distributed between bunkers beneath it; the sludge collected in the scrubber is removed periodically.

A pneumatic pressure system for the transfer of bulk loads from a bin at the ground level into a number of bins arranged at a certain height is illustrated in Fig. 15.12. The material is charged by a screw feeder into a pressure pipeline, is raised by the air stream into a branched pipeline at the top and distributed between the bins; the separated air is filtered and discharged into the atmosphere. The material from the top bins is loaded into transport vehicles by means of air-activated chutes.

A pneumatic vacuum-pressure system for ash removal in a large boiler plant is shown in Fig. 15.13. Ash is sucked off from ash collectors *1* with valves *2* into vacuum pipelines *3* and carried to a separator and a feeder of the pressure system *4* and then through

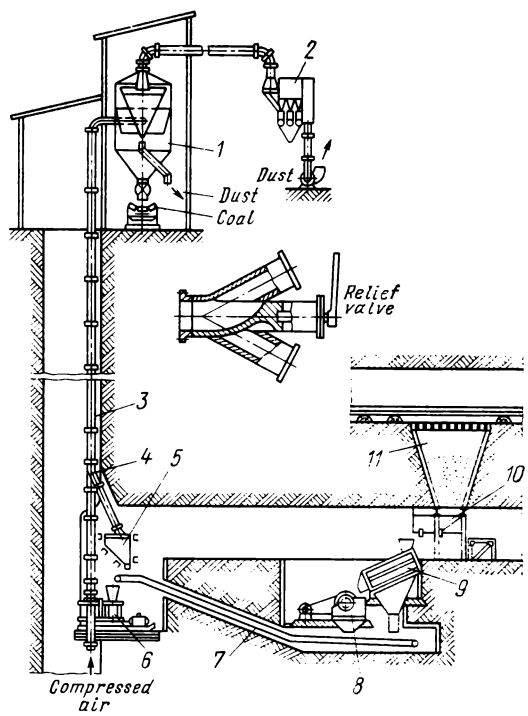


Fig. 15.14. Pressure-type pneumatic transport system for lifting coal from colliery to the surface

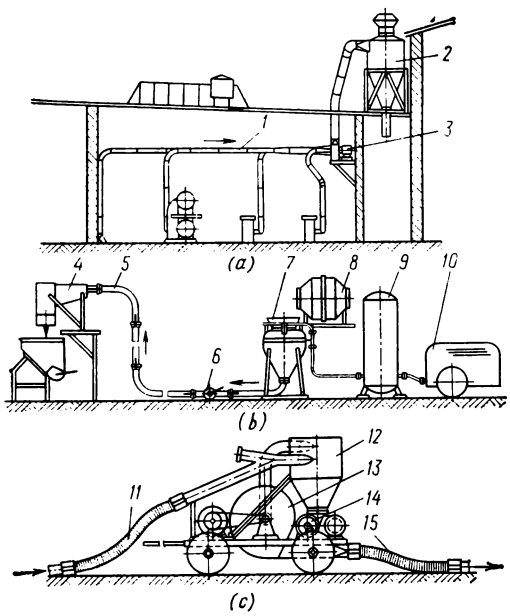


Fig. 15.15. Light-type pneumatic transport systems

(a) vacuum-type; (b) pressure-type; (c) combined-type, mobile; *1*—conveying pipeline; *2*—separator; *3*—fan; *4*—cyclone; *5*—pressure pipeline; *6*—gate; *7*—concrete mixture tank; *8*—concrete mixer; *9*—air collector; *10*—mobile compressor; *11*—suction pipeline; *12*—cyclone; *13*—fan; *14*—separator gate; *15*—pressure pipeline

a pressure pipeline *5* into large-volume collecting bins *6*. Flight feeders *7* transfer the ash from these bins into a barge *8*, railway car or collecting bin.

Figure 15.14 shows a system for pneumatic lifting of coal from a colliery to the surface. Coal from hopper *11* is transferred by a feeder *10* onto a screen *9*. Fine fractions passing through the screen fall directly onto a belt conveyer *7*, while screened-off larger lumps are crushed in a coal mill and then charged onto the belt conveyer. Coal is discharged from the conveyer into the funnel of a feeder *6* which introduces the coal flow into a working pipeline *3* in which air is compressed by a blower arranged at the bottom. The pressure pipeline has a relief valve *4* (shown separately in the figure), through which the coal in the vertical pipeline can be dropped into a bin *5* in the case of an emergency. The coal lifted to the surface is separated from air in a separator *1*; the separated air is additionally cleaned in a filter *2* and discharged into the atmosphere.

Figure 15.15 shows three schemes of light-type pneumatic transport systems: a vacuum system for removal of chips and sawdust in a woodworking shop (Fig. 15.15*a*); a pressure system for conveying of concrete mixtures and mortars on a construction site (Fig. 15.15*b*); and a mobile (on undercarriage) vacuum-pressure installation for handling grain and other non-dusting materials (Fig. 15.15*c*), which is provided with flexible suction and discharging hoses, a separator with drum-type valve and a fan to form an air stream.

15.2.3. Mechanical Equipment of Pneumatic Transport Systems

The main equipment of pneumatic transport systems includes pipelines with fittings, air blowers, feeders, suction pipes, jet devices, nozzles, separators, and filters.

Feeders. The feeders, as those of hydraulic systems, can be divided into intermittent box-type feeders which sluice the material, and continuous mechanical feeders which forcedly introduce the material into a pipeline. Besides, the material can be introduced into a pneumatic pipeline by suction-type jet devices.

An example of a two-box automatic feeder is the stowing machine employed in mines for filling in worked-out areas. The machine (Fig. 15.16*a*) has a receiving funnel and two vertically arranged chambers with successively operated valves. A bladed wheel at the bottom of the lower chamber scrapes the material to the inlet hole of the working pipeline. Compressed air is fed into the upper and also into the lower chamber to equalize the pressure before opening the lower valve. When passing from the lower chamber into the pipeline, compressed air entrains the material.

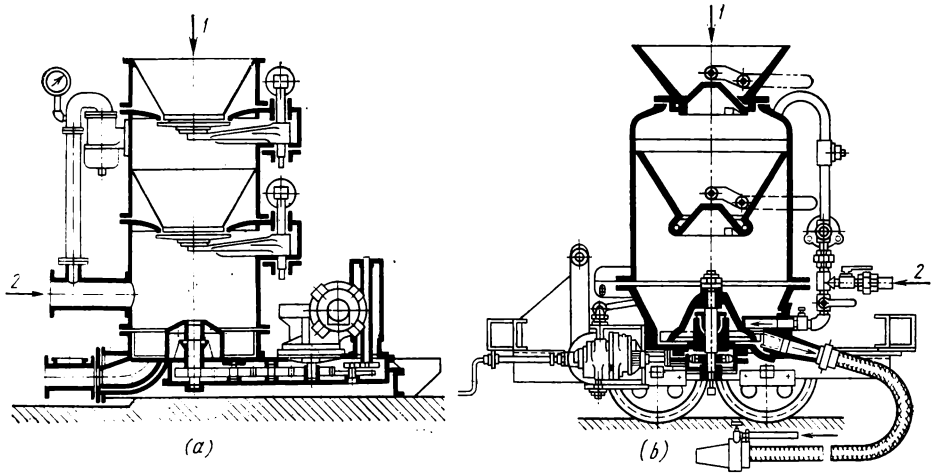


Fig. 15.16. Box-type feeders of a **pneumatic stowing** machine
(*a*) stationary machine; (*b*) mobile machine; *1*—**bulk load**; *2*—compressed air

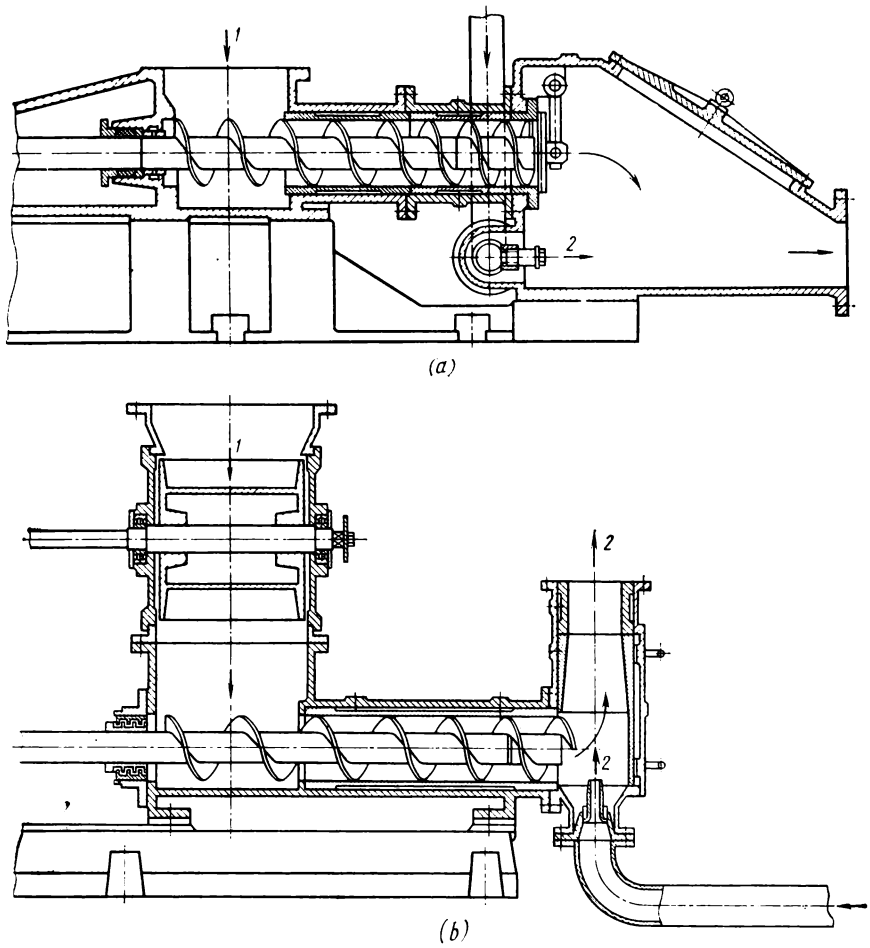


Fig. **15.17.** Screw feeders for pulverized and lumpy loads
1—**bulk** load; *2*—compressed air

A mobile feeder of this type (Fig. 15.16*b*) is convenient in cases when the loading place must be changed frequently.

Two types of screw feeders for attachment to horizontal and vertical pipelines ar illustrated in Fig. 15.17. The screws have a variable (decreasing) lead, so that the material is compacted and does not let air penetrate back into the feeder. In view of intensive wear, the feeder housing has a cylindrical insert which can be replaced easily together with the screw.

In the feeder illustrated in Fig. 15.17*a*, the material is pushed through by the screw and falls to the chamber bottom, where it breaks off (if compacted) and is entrained by the compressed air stream ejected from a nozzle on the back wall of the chamber. As the material emerges from the screw casing, it opens a tilting valve. When the screw rotates empty, the valve closes and prevents air penetration through the screw.

The feeder shown in Fig. 15.17*b* has a vertically arranged air nozzle, so that the air stream catches up the material in a vertical pipe. The feeder has no nonreturn valve, but the charging hole is closed by a drum feeder wheel which regulates the quantity of material supplied to the screw.

Separators. The transported bulk load is separated from the carrying air at the destination point, and in combined vacuum-pressure systems, also at an intermediate point. A separator is essentially a vessel of a larger cross section than the pipeline, so that the speed of the entering air stream decreases here and the suspended material falls to the separator bottom and can then be discharged through a gate. For more efficient separation of material particles, the inside surface of a separator may have guide blades made from steel strips, so that the air stream has to change its direction, which is favourable for the separation of solid particles from it. A separator of this type is

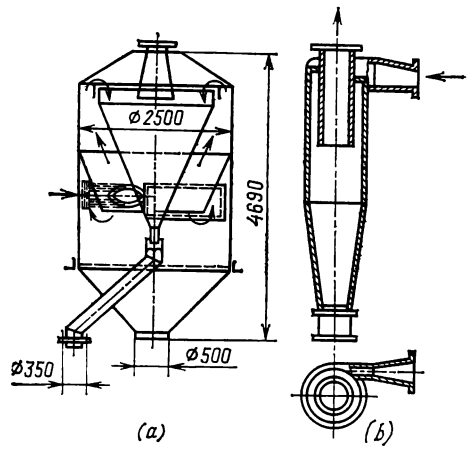


Fig. 15.18. Separators
(*a*) with internal guides; **(*b*) cyclone type**

shown in Fig. 15.18*a*. The main portion of solid particles collects on the separator bottom and finer particles are separated inside the central cone as the air stream moves to the exit hole at the separator top.

Cyclone separators for pulverized (or dust-containing) materials are based on utilizing the centrifugal force which appears when the air stream is caused to rotate inside a circular separator (Fig. 15.18*b*). A mixture of air with solid particles is admitted into a cyclone separator tangentially and begins to rotate under the centrifugal force at the separator walls. Due to the forces of friction on the walls, the speed of solid particles decreases sharply. The cleaned air is sucked off through a hole at the top and the solid material is discharged through the bottom.

Air blowers. Pneumatic transport systems employ centrifugal or piston-type air blowers, depending on the required throughput capacity and pressure and particular operating conditions. Both types of blowers have many varieties. Centrifugal machines can be divided into fans and turboblowers (various types) and piston-type machines, into rotary (i.e. with rotational motion of the working member) and reciprocating.

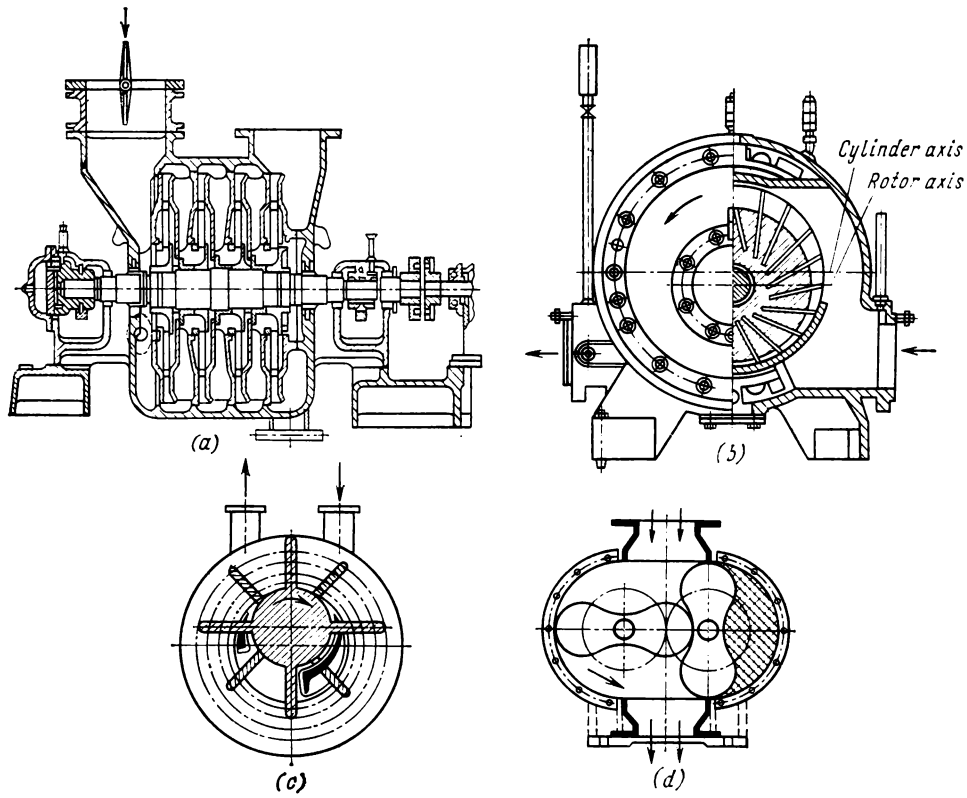


Fig. 15.19. Air blowers
(*a*) turboblower with four impellers; (*b*) rotary blower (compressor) with sliding vanes; (c) water-packed ring blower; (*d*) rotary positive (abutment) blower

Figure 15.19*a* shows the cross-sectional view of a centrifugal turboblower which is often employed in pneumatic vacuum transport systems with a high rarefaction. The shaft of the machine carries four impellers, with air moving successively around them. The rotational speed of the shaft is rather high (5000-7000 rpm).

A pneumatic liquid-packed rotary pump is shown schematically in Fig. 15.19*c*. The pump shaft with radial blades is mounted eccentrically in the cylindrical 'housing. In rotation, the rotor entrains water that forms a water ring of a practically equal thickness at the housing cylindrical walls. Thus, the free spaces for air between the blades turn out to be variable, so that air can be sucked in from an inlet hole in the end wall of the housing at one side of the rotor and discharged through another hole provided in the same wall at the other side of the rotor.

Water-packed rotary pumps are simple in design and can be protected reliably against overheating by adding fresh water. Water exchange also removes solid particles from the pump. A drawback of this type of pump is a high power loss due to friction of water on the walls of housing, and therefore, a low efficiency (0.4-0.45).

The design of a rotary swash-plate pump is clear from Fig. 15.19*b*. The rotor shaft is mounted eccentrically in the pump housing and has radial or (as shown) oblique slots in which steel or plastic blades can slide freely. In rotation, the external edges of blades slide

on guide rings in the housing. As in the previous type, air is sucked into variable-volume spaces between the blades at one side and is discharged at the other side of the rotor.

A rotary abutment blower (Fig. 15.19*d*) has two rotors in the form of an 'eight' which rotate in opposite directions. Air is sucked in at one (top) side of the housing and discharged at the other (bottom) side. Air is compressed in side spaces between the rotors and housing (shown hatched). A drawback of this type of blowers is that air can penetrate through the gaps between the rotors and housing and between the rotors, this effect being greater at higher pressure and larger wear of a blower.

Thus, among the four types of air blowers described above, the first type operates by the centrifugal principle where the kinetic energy of an air stream is transformed into potential energy of pressure, while the other three types function by the principle of displacement of an air volume by working members.

15.2.4. Automatic Control of Pneumatic Transport Systems

As with hydraulic systems, the automatic control of pneumatic transport systems has an object to maintain the specified operating conditions by regulating such characteristics as the air pressure in pipelines, the speed of air stream, the rate of supply of bulk material into the pipeline (by controlling the operation of feeders) and the rate of air supply (in particular, by controlling the rotational speed of the blower). In addition, the systems are provided with automatic locking means (switches, valves, feeders), level gauges in bunkers, etc.

As an example, let us describe a direct-action pneumatic controller for regulating the air pressure in the working pipeline (Fig. 15.20). The functions of measuring the controlled parameter and of displacing a control valve are performed by two membranes: a measuring membrane (which measures the pressure in a pipeline) and a power membrane which acts as a direct drive for the control valve. If the pressure has deviated from the set value, membrane *6* (which is acted upon by the air pressure at one side and by the force of a spring *5* at the other) and the associated slide valve *7* are displaced and change the pressure in the space *3* under the power membrane *2*. Under this action, the power membrane displaces a control valve *1* until the varying pressure of air before the valve becomes equal to the air pressure under it. The pressure controller is adjusted by means of a damper *8* and by changing the stiffness of spring *5* with screw *4*.

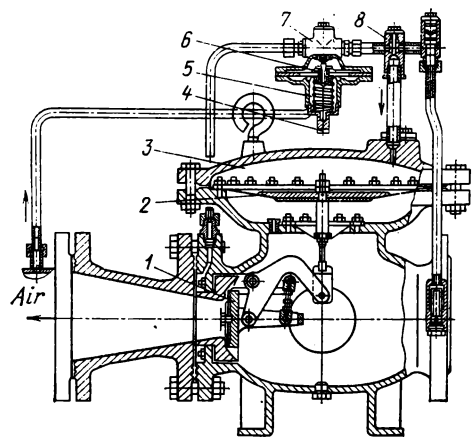


Fig. 15.20. Direct-action pneumatic pressure controller

15.2.5. Special-type Pneumatic Transport Systems

Special types of pneumatic systems include air-activated chutes and pipelines for conveying of pulverized materials and systems in which small articles or small quantities of bulk loads are carried in cartridge containers (pneumatic tubes).

Air-activated pneumatic pipelines. When some pulverized (fine-disperse)

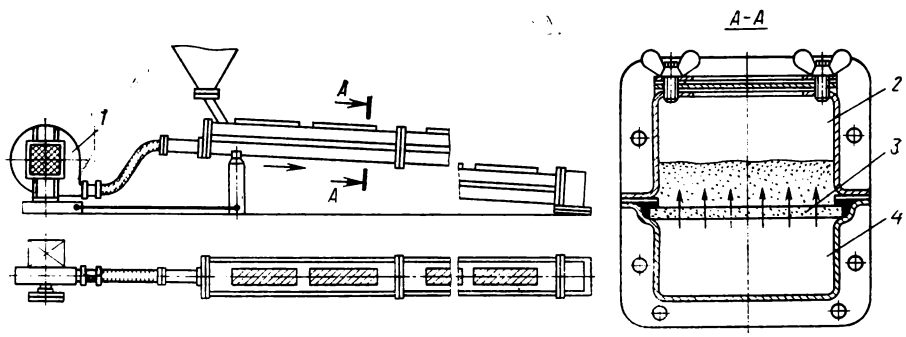


Fig. 15.21. Air-activated chute
1—fan; *2*—upper compartment for bulk material; *3*—porous partition; *4*—lower compartment for air

materials, such as cement, dry pulverized coal or flour, are saturated with air (aerated), the coefficient of internal friction in them drops down sharply and they acquire a high flowability close to that of liquids. Air-activated bulk materials can move in chutes by gravity even at a very slight decline (2.5-3°). In pressure pipelines, they fill in the entire cross section and can be moved efficiently. In this method, the concentration of mixture may be rather high, which makes it possible to decrease the flow rate of compressed air and the pipeline diameter compared with the conventional method of pneumatic transport.

Figure 15.21 shows schematically an air-activated chute. It consists of two closed parallel sections arranged one above the other and separated by a porous partition.

The material is charged into the upper compartment, and compressed air is blown by a fan into the lower compartment to form a pressure slightly above the atmospheric. The material is discharged at the chute end and the air passes out through cloth filters provided in the upper compartment cover. The partition is made from porous ceramic plates or a number of layers of a strong canvas, belting or other fabric and should be evenly permeable for air all over its surface, so that air will pass through it in very fine streaks.

An example of an air-activated pneumatic system is a cement reloader employed in civil engineering. Cement is delivered by a cement truck (Fig. 15.22*a*) in a tank *2* having on its bottom a chute consisting of two inclined porous plates *4*. A small blower *1* on the truck chassis supplies compressed air into the space under the chute and the material is discharged through a pipe connection *3*.

The receiving installation (Fig. 15.22*b*) has a vertical cylindrical tank *7*. Cement is transported from the truck through pipeline *6* into the top of the tank and air passes from the tank through a filter *5*. The material is then fed from the tank through a pressure pipeline *9* into a measuring bin *8* and then into a concrete mixer *10*. In order to activate the material in the pressure pipeline, the lower portion of the pipeline has inside a number of tubes made of an elastic air-permeable fabric. The tubes are arranged at a certain distance from one another and their farther ends are closed with plugs. As compressed air is blown into the tubes, it activates the cement carried through the pipeline, which allows the conveying process to be performed with a high concentration of cement in the air-cement mixture. Pneumatic conveying with high concentrations of solid in the air mixture has certain perspectives of application not only with pulverized, but also with fine- and large-size granular and pelletized materials.

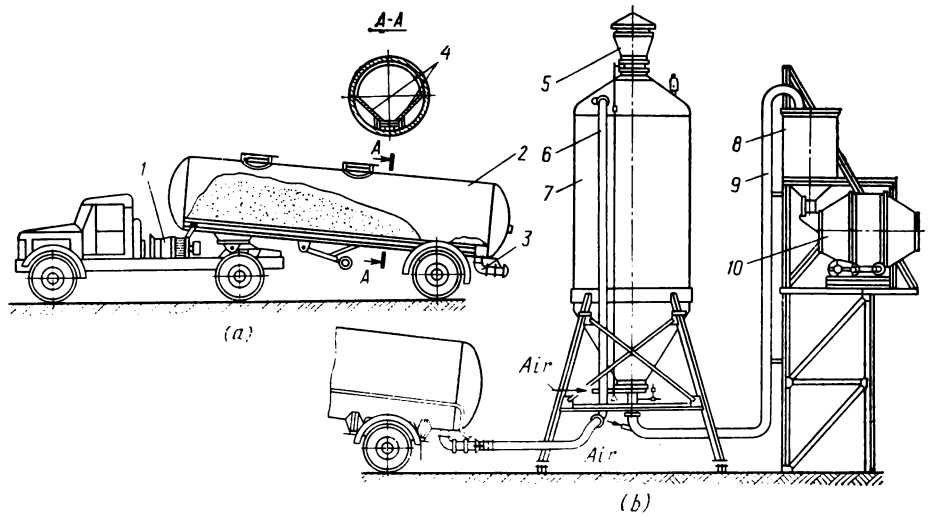


Fig. 15.22. Cement-reloading installation

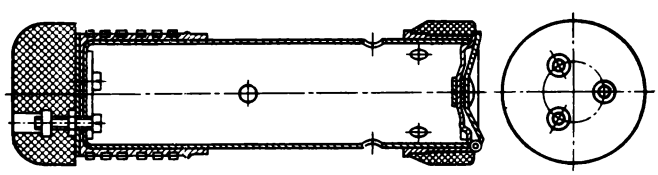


Fig. 15.23. Pneumatic mail cartridge container

Pneumatic tube conveying. This method differs principally from the pneumatic transport methods discussed earlier. The loads to be transported (small articles, books, parcels, etc. or small quantities of bulk materials) are packed into cartridge containers of cylindrical (or less frequently, of elliptic or rectangular) shape, which are propelled in a pipeline by the static pressure of compressed air.

Pneumatic tube conveying may be employed for material transport in a building or shop or between buildings to large distances (for instance, pneumatic mail).

In recent time, certain positive results have been obtained in pilot-plant installations for transportation of bulk loads in containers through large-diameter pipelines.

A cylindrical cartridge container for pneumatic tube conveying is shown in Fig. 15.23. The cartridge body is usually made of a light-weight material, such as duralumin or plastics. The sealing rings on the cartridge ends are made of an elastic material: rubber, soft plastics, etc. The load is charged into a cartridge through a side or end cover.

15.3. Calculation of Hydraulic and Pneumatic Transport Systems

15.3.1. General Considerations

The calculation of a hydraulic or pneumatic transport system proceeds from the initial specified data on the

volume or mass throughput capacity, characteristics of the load (density, particle or lump size, etc.), the length and configuration of the conveying path and has to determine the required speed of the carrier fluid (water or air), its flow rate, the diameter of pipeline, the resistances in various sections of the pipeline and the required head or pressure to overcome them, and the power of pumps or blowers. The characteristics of the carrier flow to ensure steady transportation of bulk materials, depending on their particle size, shape and density, are usually determined on the basis of the available experimental data. For that reason, some calculation formulae and recommendations in the specialist literature are empirical or semi-empirical. Since such data would occupy much place, we give below only a brief description of the method for rough calculations and some reference data.

The size of particles and the density of bulk material are of prime importance for determining the speed and head or pressure of the carrier fluid. Bulk loads can be divided into three groups by their particle size: (1) lumpy (with lump size above 40 mm) and granular: large-size (6-40 mm) and small-size (2-6 mm); (2) coarse-disperse (like sand), with particles of a size 0.15-2 mm; and (3) fine-disperse (dusty), with particle size less than 0.15 mm. These groups can be characterized by different hydromechanical processes of their motion in a carrier fluid flow.

Many bulk materials are mixtures of two or more such groups, i.e. may contain, for instance, lumps and large grains together with coarse- or fine-disperse particles. In such cases, the presence of fine particles in a material improves the conveying process of larger ones. In horizontal pipelines, fine particles usually are carried in the suspended state, while medium- and large-size particles perform short jumps in the suspended state and often touch the bottom of the pipeline; large lumps in hydraulic systems often move only by sliding along the pipe bottom. Pulverized bulk materials can be transported in pneumatic systems with a high concentration in the air flow (air-activated flows).

Material is moved in horizontal pipelines in the suspended or partially suspended state owing to turbulent motion of the carrier fluid with a certain velocity gradient (i.e. with its velocity varying from a minimum at the pipe walls to the highest value in the centre of the cross section). The forces which maintain solid particles in turbulent motion are functions of the longitudinal velocity of the flow and they increase with an increase of this velocity. For that reason, the velocity of the carrier fluid flow in pipes should, in the general case, be above a certain limit which is the higher, the greater the size and density of solid particles.

A convenient criterion for establishing the flow velocity is the critical velocity v_{cr}, i.e. the lowest velocity at which the material does not accumulate and does not block the pipeline. In pneumatic conveying of air-activated loads, the critical velocity is actually the effective velocity v_{ef} at which the load is properly saturated with air. In inclined pipelines, the critical velocity is determined by considering additionally the following factors: in hydraulic conveying, the rate of settling of particles or lumps in water (which is sometimes called 'fall velocity') and in pneumatic conveying, the soaring velocity of particles in an ascending air flow (in a vertical pipeline). The fall velocity is usually determined by the time during which a solid particle passes a definite path length when moving uniformly under the gravity force in a water-filled vessel. The soaring velocity of a particle in an air flow is equal to the air velocity in a vertical pipe at which the particle is maintained by the stream and remains in place (actually, it performs small oscillations about a certain equilibrium point).

For efficient transportation of a material, the flow velocity v must be 1.1-1.2 times the critical velocity v_{cr} for the largest lumps. On the other hand, it should not be excessively high, otherwise it would involve excessive power losses, increased wear of the pipeline, and crushing of the transported material.

Further, for normal conveying, the flow rate of water or air should be in a definite proportion relative to that of the transported material; it determines the concentration (thickness) of the mixture and is usually selected according to the available practical data.

It should be distinguished between the volume concentration and the mass concentration of a mixture. The former is usually employed in calculations of hydraulic transport systems, and the latter, for pneumatic systems.

The volume concentration of a hydraulic mixture is usually understood as the ratio of the volume throughput capacity V (m³/h) of a system to the flow rate of the hydraulic mixture V_h (m³/h) for the same time:

$$s = V/V_h = V/(V + V_w) = (\rho_l - \rho_w)/(\rho - \rho_w) < 1 \qquad (15.1)$$

where ρ_l, ρ_w and ρ are the densities of the transported load, water and hydraulic mixture, t/m³, and V_w is the flow rate of water, m³/h.

The density of the load

$$\rho_l = (V\rho + V_w\rho_w)/(V + V_w) \qquad (15.2)$$

The mass concentration of hydraulic mixture is the ratio of the mass throughput capacity Q (t/h) of the system to the mass flow rate of water, $V_w\rho_w$ (t/h), for the same time

$$\mu = Q/(V_w\rho_w) \qquad (15.3)$$

Having chosen, according to the experimental or practical data, the mixture concentration and pipeline diameter, one can determine the velocity of the mixture v_{cr}. It should then be checked that the flow rate of hydraulic or air mixture satisfies the condition: $v \geqslant v_{cr}$.

With the characteristics of the mixture flow (velocity and concentration) and the pipeline diameter being known, it is possible to calculate the resistances to motion. They include resistances to lifting of the mixture in vertical and inclined sections of the pipeline and hydraulic losses along the entire pipeline and in local sections (turns, branches, etc.). Local 'concentrated' resistances in pneumatic systems are equated to linear resistances of an equivalent length, which together with the actual length of the pipeline give its equivalent (resolved) length L_r. For hydraulic systems, local resistances are considered by introducing a coefficient 1.05-1.1 to the total pressure head.

By the values of the required pressure head and throughput capacity, one can select a particular water pump or air blower, after which the power of the drive motors can be calculated.

The calculation schemes for hydraulic (pressure and gravity-type) and pneumatic transport systems will be discussed separately in the following sections.

15.3.2. Calculation of Pressure-type Hydraulic Systems

The calculation of horizontal pipelines is made by different formulae, depending on whether the material (or mixture) to be transported is fine-disperse, coarse-disperse or lumpy.

1. For hydraulic systems for conveying of fine-disperse bulk loads (with particle size less than 0.2 mm) in a turbulent flow, the critical velocity (m/s) is proportional to the pipe diameter D and the square root of the particle density ρ_s

$$v_{cr} = n\sqrt{agD} \qquad (15.4)$$

where n = 1-1.5 is an experimental coefficient to consider the effect of mixture intermixing; and $a = (\rho_s - \rho_w)/\rho_w$ is

the ratio of the densities of load particles and carrier fluid (water).

Fine-disperse bulk materials are transported with a high concentration in the mixture, usually not less than $s = 0.2\text{-}0.25$ or even up to 0.5.

The chosen pipe diameter should be checked to satisfy the condition

$$v = 4V_h/(3600\pi D^2) \geqslant v_{cr} \quad (15.5)$$

where V_h is the flow rate of the hydraulic mixture, m^3/h.

The unit head loss (m/m) for the moving mixture

$$H' = k_1 H_0 (1 + as) \quad (15.6)$$

where $k_1 = 1.1\text{-}1.5$ is a coefficient to consider the intermixing effect in the mixture, and H_0 is the unit head loss for a flow of pure water moving at the same velocity as the hydraulic mixture, m/m

$$H_0 = \zeta v^2/(Dg) \quad (15.7)$$

where ζ is the coefficient of hydraulic resistance of a smooth pipe (see in the courses on hydraulics).

If the pipeline has vertical sections of a lifting height L_l, the required pressure head will be higher by the static head of lifting H_l. If the mixture moves downwards, the required head will be accordingly smaller. Therefore

$$H_l = \pm L_l \quad (15.8)$$

In addition to these distributed losses, there are usually additional local losses H_{lr} which are usually taken equal to 5% of the total losses in straight sections.

2. For hydraulic systems to convey lumpy and granular bulk materials with a particle size more than 2 mm, the critical velocity should be determined by the formula

$$v_{cr} = C_1 \sqrt{fagsD} \quad (15.9)$$

where $C_1 = 8.5\text{-}9.5$ is an experimental coefficient and f is the generalized coefficient of material friction on the pipe bottom; its values for various bulk materials are given in Table 15.1.

The critical velocities of the hydraulic mixture for some groups of bulk materials and different values of pipe diameter are given in Table 15.2.

The unit head loss for a moving mixture is the sum of the losses for the motion of pure water and additional losses caused by the presence of material particles and lumps in it, i.e.

$$H' = H_0 + fas \quad (15.10)$$

For efficient transportation of a lumpy material without accumulation in the pipeline, the largest size of its lumps should not exceed 1/3 of the pipe diameter and the highest concentration should be not more than $s = 0.2\text{-}0.25$.

Table 15.1. Generalized Coefficients of Friction

Material	Lumps with edges	
	sharp	rounded-off
Coal	0.3-0.2	0.25-0.15
Anthracite	0.15	0.1
Sandstone	0.55-0.5	0.5-0.45
Limestone	0.45-0.4	0.4-0.35
Oil shale	0.4-0.35	0.35-0.3
Gravel	—	0.4-0.35
Ore	0.65-0.75	0.6

Table 15.2. Critical Velocities (m/s) of Hydraulic Mixture

Pipeline diameter, mm	Material				
	with clayey particles	sand particles with clay admixture	sand and gravel (50%) with low content of clay particles	gravel, crushed stone	ungraded coal
200	1.6	1.9	2.4	3.0	2.0
300	1.8	2.1	2.9	3.6	2.5
400	2.2	2.4	3.4	4.3	3.0
500	2.5	3.0	3.8	4.8	3.3
600	2.7	3.2	4.1	5.3	3.6

Hydraulic systems for transportation of coarse-disperse bulk loads and their mixtures are calculated in a similar way.

Having found the total required pressure head (m)

$$H_r = H_l + H_{lr} \qquad (15.11)$$

one can choose a proper pump for the given throughput capacity V (m³/h) of the system and determine the required power of its motor (kW)

$$N = k_s H_r V \rho_l / (367\eta) \qquad (15.12)$$

where $k_s = 1.1\text{-}1.2$ is a safety factor and $\eta = 0.7\text{-}0.9$ is the efficiency of the pump.

15.3.3. Calculation of Gravity-type Hydraulic Systems

The calculation of hydraulic systems in which the load moves by gravity in pulp conduits, open chutes, channels proceeds from the given throughput capacity for a compact (unloosened) mass of the material and its characteristics (kind and particle size) and determines the parameters of the carrying flow (relative quantity and speed of water), the angles of decline, and cross-sectional dimensions of chutes.

The principal formula for the calculation (Chezy formula) correlates the stream velocity v (m/s), hydraulic radius R (m), and chute slope i

$$v = C\sqrt{Ri} \qquad (15.13)$$

or

$$i = v^2/(C^2R) \qquad (15.13')$$

where the Chezy coefficient C depends on the roughness of chute surface and on the hydraulic radius R. The hydraulic radius, or the ratio of the cross-sectional area of the stream to the wetted perimeter, is determined by the form of the open chute, depending on chute dimensions. For instance, for a semicircular chute of a diameter D, filled with hydraulic mixture, the hydraulic radius

$$R = \frac{\pi D^2}{4 \times 2} : \frac{\pi D}{2} = \frac{D}{4} \qquad (15.14)$$

For an open chute of rectangular cross section with the width B and the stream depth h, the hydraulic radius

$$R = Bh/(B + 2h) \qquad (15.15)$$

For the recommended ratio $B/h = 3\text{-}4$.

$$R = (0.6\text{-}0.66)\, h$$

The hydraulic radius R for trapezoidal cross sections of chutes can be found in a similar way.

The coefficients C for various values of the relative roughness λ of the surface of open chutes and of the hydraulic radius R may be taken from Table 15.3.

The values of relative roughness λ for various kinds of surface of open pulp conduits (chutes, channels) are as follows:

Carefully planed, well fitted boards; cement plaster	0.010
Unplaned well fitted boards; clean metal pipes; smooth concrete	0.012
Brickwork; surface-tooled stone	0.015
Rubblestone; chanells in dense gravel, loess, compacted soil, covered with silt film	0.020

Table 15.3. Coefficients C for Different Values of Hydraulic Radius R and Relative Roughness λ

λ	R, m				
	0.1	0.2	0.3	0.4	0.5
0.010	70	75	80	90	95
0.012	60	65	70	75	78
0.015	45	50	55	58	60
0.020	30	35	38	40	43

The approximate minimal slopes of pulp conduits for various kinds of

Table 15.4. Approximate Values of Minimal Slope for Open Pulp Conduits

Transported material	Chutes and concrete channels	Earth channels
Clay	0.015-0.0625	0.02-0.03
Sand:		
fine-size	0.025-0.030	0.03-0.04
medium-size	0.030-0.035	0.04-0.05
large-size	0.035-0.050	0.05-0.06
Gravel	0.050-0.10	—

transported materials are given in Table 15.4.

It may be recommended to calculate chutes and channels in the following order. For a given volume flow rate of transported load V' (m^3/h) and the loosening coefficient k_l ($k_l > 1$), one determines the rated volume flow rate $V = V'/k_l$. Then an appropriate concentration of the mixture, s, is chosen along the recommendations given earlier and the required throughput capacity for the hydraulic mixture, V_h (m^3/h), is calculated. After that, the mixture velocity v is selected at a level equal to or greater than the critical velocity. Finally, for the values of V_h and v thus found, it is possible to determine the geometrical dimensions of the chute cross section and its hydraulic radius R.

Having then selected a particular kind of surface lining and its relative roughness, one calculates the Chezy coefficient C for the known values of R and λ and determines by formula (15.13′) the required slope i of the chute. The value of i should be checked to comply with the data of Table 15.4.

In many practical cases, the slopes of chutes are determined by the profile of the terrain. On steeper slopes it is advisable to employ chutes with an elevated relative roughness of the surface, and vice versa.

15.3.4. Calculation of Pneumatic Transport Systems

As with hydraulic systems, the initial data for the calculation of a pneumatic transport system are the throughput capacity (Q, t/h, or V, m^3/h), the length and configuration of the pipeline, and the physico-mechanical properties of transported bulk loads which determine the coefficients in calculation formulae.

Pneumatic transport may be performed in various systems with different mass concentrations of the air mixture.

In negative-pressure (fan-driven) systems, the mass concentration of the air mixture does not usually exceed $\mu = 1$ or may reach $\mu = 5$ at a high vacuum. In pressure systems, the mass concentration is taken within a range of 8 to 25 depending on the particular conditions; for air-activated material flows, $\mu = 60\text{-}150$.

The choice of μ is dictated by the degree of looseness and slumpiness of the material, its tendency to form lumps, the presence of moisture (clay impurities), the characteristics of the pipeline path (number of turns, lifts, etc.), the capacity of the material for activation by air, etc. On the other hand, it should be considered that the unit power consumption of a pneumatic system increases with decreasing concentration of the air mixture. For that reason, too low concentrations are not recommended.

The size of particles is essential mainly for selecting the pipe diameter to avoid blocking of the pipeline ($D \geqslant \geqslant 3a$, where a is the size of largest lumps). In that connection, the formulae given below are suitable for rough calculations for a wide range of particle size of bulk loads (except for large-size lumpy ones).

The speed of air varies along the length of a pipeline: in the vacuum and pressure systems, it increases from the starting point of transportation to the end point. Indeed, in a vacuum

system, the degree of air rarefaction increases in the direction of motion; in a pressure system, the pressure gradually decreases to the atmospheric, i.e. in both cases the density of air decreases, and therefore, the volume of air (for a constant flow rate) increases. The capacity of the air stream for carrying solid particles is then directly proportional to the second power of velocity and inversely proportional to the density of the fluid, and therefore, increases from the beginning to the end of a pipeline. This is why the calculation is carried out for both types of systems for the initial section of the pipeline.

The critical velocity (m/s) is proportional to the square root of mass concentration, load density and pipe diameter:

$$v_{cr} = C_2 \sqrt{\mu a g D} \qquad (15.16)$$

where C_2 is an empirical constant and μ is the mass concentration of the air mixture [see formula (15.2)].

For easily activated pulverized materials, it is possible to take $C_2 = 0.1$; for materials liable to slumping or containing moisture and for complicated pipeline paths, a higher value of C_2 should be assumed (up to 0.25); for granular materials, C_2 may be up to 0.3, and for lumpy ones, up to 0.35 (if moisture content is up to 4%).

As in calculations of hydraulic systems, the chosen pipe diameter should be checked to satisfy the condition $v \geqslant v_{cr}$.

The resistance to motion of an air mitxure in pipeline, and therefore, the pressure (MPa) at the beginning of a pressure pipeline are higher at higher values of the resolved pipeline length L_r, mixture concentration, and resistance to motion of pure air:

$$p_p = 0.1 p_0 (1 + C_3 \mu a g D / v^2) \qquad (15.17)$$

where C_3 is an experimental constant, C_3 = 01-0.075 (lower values are taken for loads of higher density).

The pressure loss (MPa) in isothermal motion of pure air is determined by the formula:

$$p_0 = 0.1 \left(\sqrt{\frac{Q_a^2 R T \lambda L_r}{F^2 g D \times 10^8} + \frac{p_e^2}{10^2}} - 1 \right) \qquad (15.18)$$

where Q_a is the flow rate or air, kg/s; $R = 29.3$ is the universal gas constant; T is the absolute temperature of the surroundings, °C; λ is the coefficient of resistance (for pipes with a diameter of 150 mm, 175 mm, and 200 mm, λ may be taken respectively equal to 0.016-0.02, 0.015-0.018, and 0.014-0.016); L_r is the resolved pipeline length, i.e. the total length in which local resistances are replaced by equivalent straight pipe sections at $\mu \geqslant 10$ (see Table 15.5); F is the cross-sectional area of pipe, m²; and $p_e = 0.105$ MPa is the pressure at the end of the pipeline.

Table 15.5. Lengths (m) of Equivalent Sections

Kind of material	Angle of turn of a radius 0.7-1 m, deg					
	8-12	15	20	30	45	90
Pulverized	—	—	2	4	6	10
Granular	—	—	3	5	8	15
Lumpy	8	10	12	15	25	45

For an isochoric flow (Δp = constant), p_0 is calculated by formula (15.7).

The pressure losses in a pipeline include the losses due to resistance in straight sections, additional losses in pipe turns, inertia losses (associated with the acceleration of the load to the working speed), and losses in vertical and inclined rising sections, p_l.

The inertia losses of pressure (MPa) can be found by the formula:

$$p_{in} = \frac{v^2}{2} \rho_0 (1 + \beta\mu) 10^{-5} \qquad (15.19)$$

where $\beta = 0.35\text{-}0.85$ is the index of relative velocity of solid particles; for pulverized loads, $\beta = 0.6\text{-}0.85$.

The pressure losses (MPa) in short rising sections of a pipeline of a height H_l (m) can be found as:

$$p_l = 10^{-3}(1 + \mu)\rho_0 g H_l \quad (15.20)$$

The total required pressure (MPa) in the conveying system

$$p = p_p + p_{in} + p_l + p_f \quad (15.21)$$

where p_f is the pressure loss in the feeder.

The air flow rate (m^3/s) at the blower discharge

$$V_a' = kV_0 \quad (15.22)$$

where $k = 1.1\text{-}1.15$ is the coefficient of air leakage in the pipeline, feeder, etc. and V_0 is the air flow rate required for conveying.

The required power (kW) of the blower motor

$$N_m = k_2 L_{bl} V_a/(1000\ \eta) \quad (15.23)$$

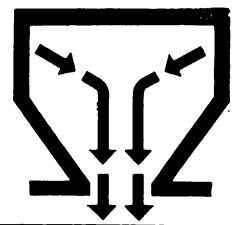
where L_{bl} is the theoretical work of the blower related to 1 m^3 of sucked-in air in isothermal compression, N m/m^3

$$L_{bl} = 2303\ p_e \log (p_p/p_e) \quad (15.24)$$

where p_p and p_e are the initial and final pressure in compression and $\eta = 0.65\text{-}0.85$ is the blower efficiency.

16

Auxiliary Conveying Equipment



16.1. Gravity-type Devices

16.1.1. Chutes and Pipes

In gravity-type conveying devices, bulk or unit loads move on a declined surface or vertically under the force of gravity.

If a load of mass m slides with a coefficient of friction f along a declined plane at an angle of slope β and on length l, with the difference of heights between the starting and end points being $h = l \sin \beta$ and the speeds of the load at these points, respectively v_s and v_f, the work of the force of gravity of the load will be spent to overcome friction and to increase the kinetic energy of the load

$$mgh = mgf\, l \cos \beta + m(v_f^2 - v_s^2)/2 \qquad (16.1)$$

For given speeds at the starting and final points, v_s and v_f, and the level difference between them h, the angle of slope β of the plane can be found from the equation

$$\tan\beta = 2ghf/(2gh + v_s^2 - v_f^2) \qquad (16.2)$$

If we know the initial speed v_s and the angle of slope β, the final speed

$$v_f = \sqrt{2gh\,(1 - f \cot \beta) + v_s^2} \qquad (16.3)$$

With $v_f = v_s$, i.e. with a constant speed of motion, $\tan \beta_0 = f$. For accelerated motion, $(v_f > v_s)$, $\beta > \beta_0$ and for decelerated motion $(v_f < v_s)$, $\beta < \beta_0$.

If a bulk load moves in a chute of a rectangular or rounded-off cross section, the coefficient of sliding friction is taken as a resolved coefficient which considers the friction of the material on the chute walls, as well on the bottom.

The speed of bulk materials in declined chutes is usually taken up to 2.5 m/s and the coefficient of filling of chutes, $\psi = 0.5$-0.7. The cross-sectional area F_0 of a chute can be determined by equation (2.7) or (2.8).

In conveyer systems, short chutes (trays) are often employed to direct the loads at loading, unloading and reloading points. In some cases, say, when transferring hot or dusty loads, closed chutes in the form of pipes of round, rectangular or shaped cross section are employed. Bifurcated and tilting chutes are used when the material should be directed to more than one point. Two examples of tilting chutes for distributing the material over a circle are illustrated in Fig. 16.1. In the design shown in Fig. 16.1*b*, the chute consists of two pipe sections connected by an articulated joint, so that the material can be distributed over an annular area whose smaller (inner) radius is obtained when the pipes are turned at the largest angle relative to

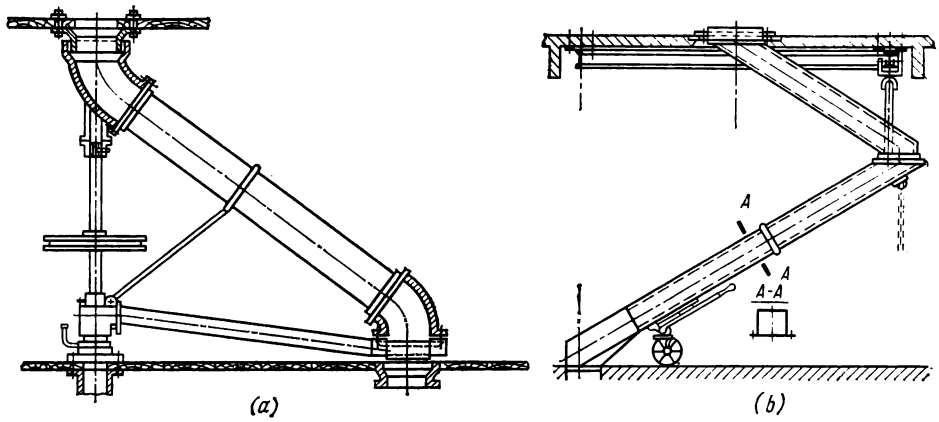


Fig. 16.1. Tilting pipe chutes

each other (as shown in the figure) and the larger (outer) radius, when they are turned in-line. The chute may be provided with a tilting drive mechanism and remote control.

Curved chutes are employed when the flow of a bulk material should be directed at an angle from the conveyer line. In that case it is essential to select a proper shape of the chute so as to minimize the resistance to motion of the material in a continuous (uninterrupted) flow.

In some cases, it is needed to combine two or more flows into one and to direct it properly or, on the contrary, to divide a flow of bulk load into two flows. This is done by means of bifurcated chutes which are assembled from properly shaped sections (Fig. 16.2).

The service life of chutes and pipes for conveying abrasive materials is sometimes increased by making them from or lining with wear-resistant materials, such as plates of chrome-nickel cast iron, manganese steel, fused basalt, etc.

16.1.2. Vertical Chutes

In vertical chutes and pipes, the load moves vertically down under the force of gravity and sometimes falls freely (for instance, filling materials can be thrown into a shaft to fill in worked-out areas, wood props can be

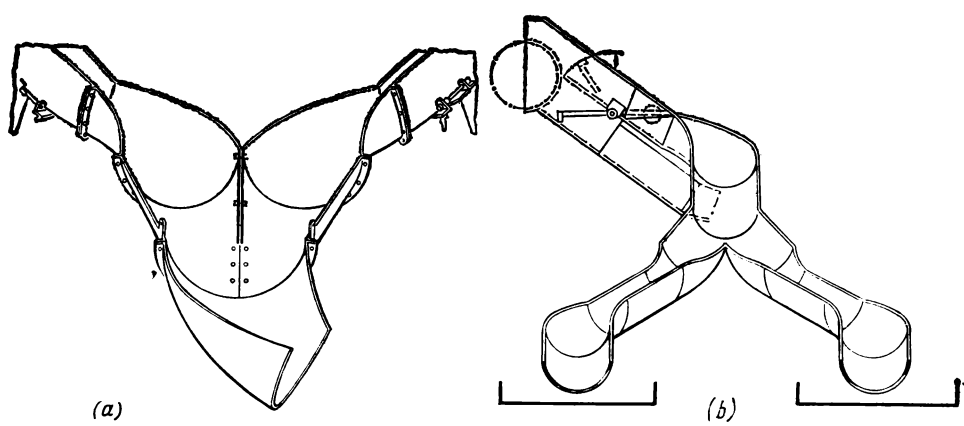


Fig. 16.2. Bifurcated chutes

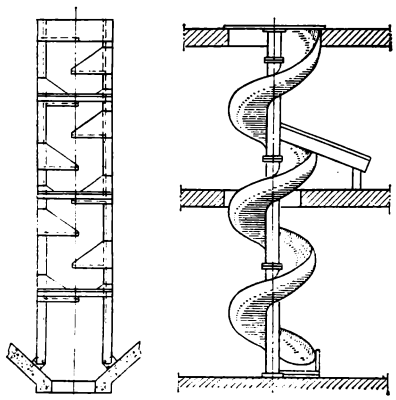


Fig. 16.3. Vertical chutes (slides)
(a) stepped; (b) spiral

dropped from a low height, etc.). In some cases, calibrated unit loads of uniform dimensions (such as boxes) can be dropped down through a properly shaped vertical pipe. The air under a falling box is compressed and is squeezed through the gap between it and pipe and thus diminishes substantially the falling speed.

The falling speed of bulk loads can be diminished by arranging a stepped (cascade) or spiral chute. A stepped vertical chute (Fig. 16.3*a*) consists of a vertical pipe with horizontal shelves attached in staggered order to its walls on the inside. The material flows from one shelf onto a next one below it and thus its falling speed is decreased. The piles of material accumulated on the shelves protect them against wear.

Brittle bulk loads and unit loads in hard (boxes) or soft (bags) packing can be delivered downwards by sliding on a spiral chute (Fig. 16.3*b*). A spiral chute may have a rectangular, trapezoidal or, most often, semicircular shape in the cross section. The chute helix has the minimal angle of slope at the periphery and the maximal, near the centre. This difference of sloping angles determines a particular property of spiral chutes, i.e. the speed of moving loads is self-controlled within definite limits. Indeed, as the speed of a increases above a certain average (for instance, due to a lower than coefficient of friction), the centri force acting on the load increases throws it to the external wall o chute where the sloping angle is sm and the load will have to move a longer path and with a higher fri and its speed thus diminishes. A speed has decreased, the centri force decreases too, and the load n onto a helical path of smaller radiu larger sloping angle, and its s will thus increase. With steady m of loads on spiral chutes, their s practically varies slightly aroun average value.

The advantages of spiral chute clude simple design, high throug capacities, and absence of moving p Their drawbacks are wear of the and loads and probable blocking o chute when the conveying condi change sharply (say, an increased sture content of the load). For reason, spiral chutes in closed pipe provided with manholes through w the chute can be cleaned from bloc up material.

A spiral chute can be used a intermediate accumulator, by cl its discharge hole. The speed of in spiral chutes is usually 2.5-3.0 and the sloping angle at the peripl of an order of 45°. Spiral chutes, straight ones, may be lined with v resistant materials.

For a material to move on a s chute at a given speed v, the slo angle and the helix lead can be rou determined in the following way. A of mass m lying on the helical su is acted upon by the components o gravity force and the force of fri of the load on the bottom and wall of the chute, and by the centri force which presses the load agains wall

$$mg \sin \alpha - mgf \cos \alpha = m (v \cos \alpha)^2 f/R_0 \quad ($$

where α is the sloping angle of the helix line of radius R_0; f is the coefficient of friction of the load on the chute; and R_0 is the radius of the helix line along which the load moves.

Hence, assuming a particular speed and having found the angle α, we can determine the helix lead

$$t = 2\pi R_0 \tan \alpha$$

16.2. Bins

16.2.1. Types and Designs

Bins and bunkers are reservoirs of a substantial volume capacity which are mounted in a common transporting and processing line and used for temporary accumulation and storage of bulk loads and for their delivery to transporting and processing equipment. Bins are charged through the open top or charging holes in the top cover and discharged usually through holes in their bottom or in the bottom portion of side walls. The material moves in a bin and flows through a discharge hole under the action of its gravity force.

As intermediate storage facilities in a line of transport and process machines, bins and bunkers ensure independent operation of these machines and the latter can run under the optimal conditions. Bins are indispensable in cases when the transport and process machines in a line operate in different time regimes (for instance, some of them periodically and others, continuously). This difference in the time regimes and the throughput capacities of the machines in a line determine the required capacity of bins. With a larger difference in time regimes (which determines the time for accumulation of a material [illegible]), and with a higher productivity of machines, bins of larger volume are required. In that connection, it is necessary to distinguished between the useful volume V_u of a bin and its [illegible] volume V_0 which are interrelated by the formula $V_u/V_0 = \psi$, where ψ is the coefficient of filling of a bin, which depends on its shape and dimensions and the method of charging.

The useful volume of a bin can be determined most simply by constructing the graph of its loading and unloading during a completed cycle of operations, for instance, during a shift, day, etc. The times in a cycle are laid off as abscissae and the volumes (m^3) of material loaded into the bin and unloaded from it, as ordinates. The largest ordinate between the loading and unloading curves determines the largest volume of material that must be accumulated in the bin, and therefore, the required useful volume of the bin.

The shape of a bin should satisfy the building requirements and should be such as to ensure complete loading and unloading without the formation of 'dead zones', i.e. zones where the material is retained and cannot flow by gravity to the discharge hole. This is mostly due to an insufficient slope of the side walls along which the material is directed to the discharge. Besides, the bin shape should be such as to prevent bridging of the material which might disturb the conditions of discharge from the bin.

Some typical geometrical shapes of bins are illustrated in Fig. 16.4. The bin usually consists of two sections: a prismatic or cylindrical upper section and a pyramidal, conical or spherical (narrowing to the discharge hole) lower section. In low bins, the upper section may be absent. Large bins are often made of an elongated shape in plan and provided with several discharge holes. In some cases, bins are divided by partitions into a number of cells or arranged side by side in two or more rows.

The common materials for bins are metals, concrete, reinforced concrete and wood. The most popular designs of bins have steel sheet walls and a riveted or welded supporting structure from rolled steel sections. Bins are

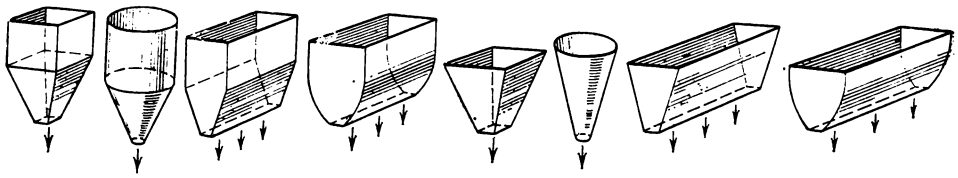


Fig. 16.4. Typical shapes of bins

usually mounted on columns or floors of buildings. The walls of bins should be smooth inside to facilitate motion of the load; bins for abrasive materials are lined with replaceable steel plates or special wear-resistant glass. Heat-insulated and heated bins are employed in cases when the delivered materials may be wet or frozen.

The discharge holes may be provided n the centre of the bottom or at one or two sides from the central line. The bin shown in Fig. 16.5*a* has a central discharge hole and two side holes. It has a parabolic shape in which the walls are subjected mainly to tensile stresses, but not bending ones, so that the bin has a lower mass than bins with straight walls. The bin is suspended by its top edges from the supporting structure.

In order to avoid construction of

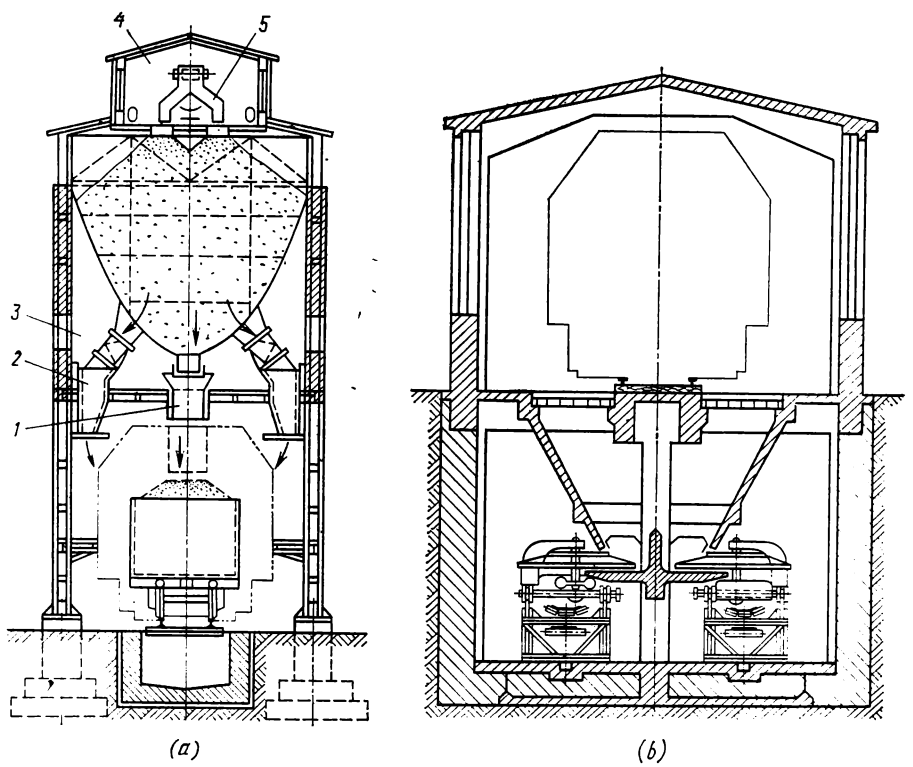


Fig. 16.5. Reloading bins

(*a*) for material reloading from a conveyer into railway cars; (*b*) for reloading from railway cars onto a conveyer; *1* and *2*—central and side discharge holes; *3*—heated space under the bin; *4*—conveyer gallery; *5*—unloading carriage of belt conveyer

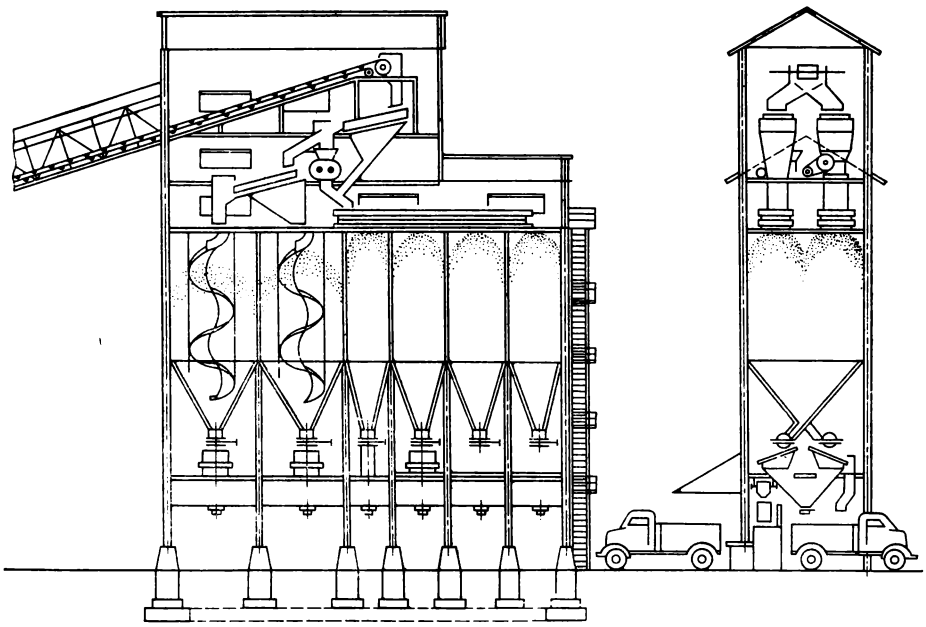
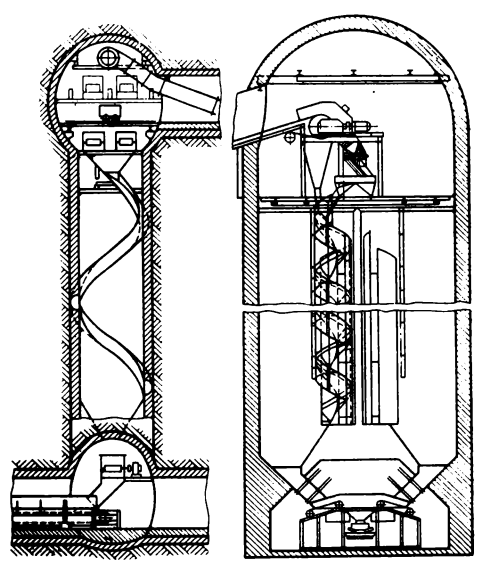


Fig. 16.6. A bin with six cells

heavy railway overpasses, bins for receiving bulk loads from railway cars are often built under the rail level (Fig. 16.5*b*). Their typical feature is that discharge holes are replaced by two slots along the entire length of a bin; the slots are protected from beneath by shelves which prevent spontaneous outflow of the material. The material is discharged on two belt conveyers by means of flight or pusher feeders moving along the discharging front (see Fig. 16.14). Bins of this type (which may be called 'slotted bins') have certain advantages when unloading should be carried out on a long front and with a high throughput capacity or when the handled materials are liable to bridging.

Figure 16.6 shows a bin which consists of six cells: two cells (on the left) for large lumpy loads and four, for small-lump materials. Two screens and a grill are mounted in the room above the bin. The material is delivered by an inclined belt conveyer and discharged onto the first screen. Fines separated by the screen fall onto a distributing conveyer mounted above the four cells on the right. Larger lumps from the



(*a*) (*b*)

Fig. 16.7. Shaft-type reloading and storage bins with spiral slides

first screen are directed either into the mill for crushing and then to the distributing conveyer or onto the second screen. The material on the second screen is divided into two lump-size classees which are loaded into the two cells at the left. Spiral chutes are suspended in both cells at the left in order to diminish crushing of the material and protect the bin gates against impacts of falling lumps. Under the bin, larger lumps are screened off from fines before being loaded into trucks.

Figure 16.7 shows high shaft bins of cylindrical shape for storage of coal before it is delivered into cars or onto conveyers. In the former case, the spiral chute is fastened on the bin walls and in the second, suspended near the bin axis. Bins of this type have a diameter of 3.5-10 m and a volume up to 3000 m^3.

16.2.2. Outflow and Bridging in Bins

The process of outflow of a bulk material from the discharge hole of a bin occurs essentially as shown in Fig. 16.8*a* and *b* (respectively with central and side position of the hole). In either case, only the vertical column of the material just above the hole comes into motion as the gate is opened, so that a cone forms on the surface of the material and, as the material flows out, its particles pour down along the sides of the cone. If the discharge hole is arranged near a vertical wall of the bin, the material slides along that wall and the cone is unsymmetrical.

By experimental data, the outflow of material through a symmetrically arranged bin hole occurs in the following manner (Fig. 16.8*c*): portion *1*, which has the shape of a cone or wedge, depending on the shape of hole, falls out first; after that the elliptical portion *2* above it falls out; then portions *3* and *4* come into motion, so that at the end of discharge only portions *5* remain in the bin and form a conical 'dead zone'.

The smallest angle of slope of bin walls in the bottom portion at which no 'dead zone' will form, but the material will flow out completely, depends on the coefficient of friction of the material on the wall (it increases with an increase of the coefficient of friction). For instance, this angle is roughly equal to 45-50° for graded coal, 60° for fine coal, and at least 65° for ore. As the material pours from the discharge hole, its remaining portions may often form a 'bridge' in the form of a dome or arc. This effect is common with materials containing large lumps (Fig. 16.8*d*), but can also occur with small-size lumpy

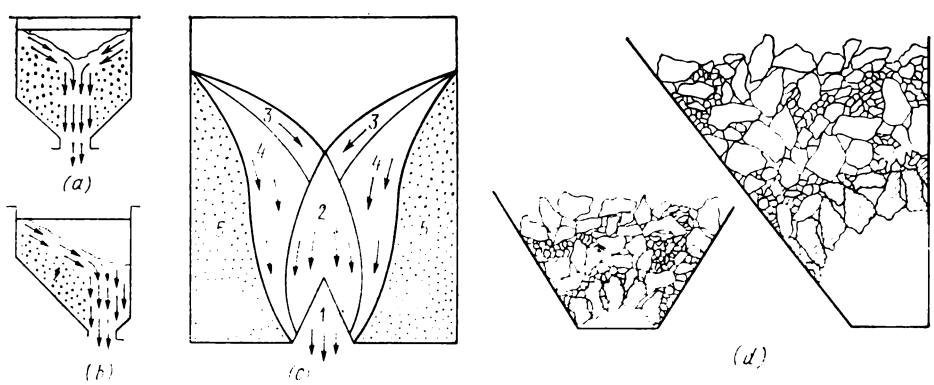


Fig. 16.8. Diagrams of material flow from bins
(*a*) flow from a central hole; (*b*) flow from a side hole; (*c*) portions of material successively involved into flow; (*d*) bridging

loads, especially those having a_t high moisture content or liable to slumping. A bridge is compressed transversely by the gravity force of the material layers above it and is supported at the ends by the inclined walls of the bin bottom or, in bins of a small cross-sectional size, even by the vertical walls.

Bridging in bins is eliminated by various hand tools and mechanical means. In the simplest case, a bridge is destroyed by pokers through special manholes in the bin walls, which have baffles at the inside to prevent fall-through of the material to the outside. Mechanical devices may be in the form of a weight suspended from a chain, or a rotating bladed wheel. In some cases, vibrators are attached to the steel walls of bins to activate the outflow of the material. Vibrators are actuated only with the discharge hole open and the feeder operating; otherwise vibration can produce an inverse effect, i.e. enhance slumping and compaction of the material in the bin. Another method is to blow in compressed air in strong intermittent jets through air tuyeres arranged somewhat below the level of probable bridging. With pulverized materials, compressed air activates then and increases the flowability. Means for combatting the bridging effect are chosen in each particular case, mainly according to the properties of the material.

16.2.3. Pressure on Walls and Gates of Bins

As is known, the hydrostatic pressure of a liquid (or of 'ideally flowing' loose material) on the walls of a vessel at a depth h from the top level of the liquid

$$p = \rho g h$$

where ρ is the density of the liquid.

For real bulk materials which possess internal friction (the friction of particles on one another), the pressure on the vessel walls is lower

$$p = \rho g h k \qquad (16.5)$$

where k is the coefficient of flowability of a bulk material

$$k = (1 - \sin \varphi)/(1 + \sin \varphi) < 1 \qquad (16.6)$$

here φ is the angle of internal friction which is roughly equal to the angle of repose of a material at rest.

Formulae (16.5) and (16.6) give approximate values of the pressure of material on bin walls for bins of a small depth, i.e. those whose depth does not exceed the cross-sectional size. For deeper bins, this relationship is disturbed mainly due to increased forces of friction of the material on bin walls, whose effect is stronger at a higher coefficient of friction and a smaller ratio of the bin cross-sectional area F to its perimeter L, i.e. a smaller hydraulic radius of the bin cross-section ($R = F/L$).

Beyond a certain depth limit, the highest horizontal pressure of the material on bin walls is not increased further, but is practically stabilized. The pressure is directly proportional to the density of the material and the hydraulic radius R and inversely proportional to the coefficient of friction of the material on bin walls, f

$$p_{max} = \rho g R/f \qquad (16.7)$$

The highest vertical pressure is greater than the highest horizontal pressure

$$q_{max} = \rho g R/(kf) \qquad (16.8)$$

Since k and f vary roughly in inverse proportion to each other (with a decrease of the angle φ, k increases and f decreases), it is assumed in practical calculations that

$$kf \approx \text{constant} = 0.2$$

Then the pressure (Pa)

$$q_{max} \approx \text{constant} = 5\rho Rg \qquad (16.9)$$

The size of discharge holes of bins should be sufficient to ensure the required throughput capacity. On the

other hand, they should not be excessively large so as to avoid the danger of fall of large masses of the material; further, large discharge holes will require heavy gates which might be difficult to operate.

For lumpy bulk loads with typical lumps of a size a, the smallest size of the discharge hole of rectangular form should be:

for bottom holes

$$b \geqslant (4.0\text{-}5.5)\, a$$

and for holes in inclined bottom walls

$$b \geqslant (4.5\text{-}6.0)\, a$$

Besides, it is essential to consider the percentage of large lumps and the properties of the material, in particular, its flowability. For small-size lumpy loads, the dimensions of bin holes are taken depending on the required throughput capacity, but not less than 300-400 mm.

16.2.4. Automatic Control of Bins

Automatic control of bins consists mainly in measuring the level of material in them and actuating the mechanical devices for material loading and unloading.

The degree of filling of bins is controlled by various level indicators (sensors), which may be mechanical (based on mechanical contact with the material), photo-electric, or radiometric.

A level gauge based on the mechanical principle has a pendulum ('float'), i.e. a pivoted rod with a hollow sphere on its end, which is mounted in the bin near the side wall (Fig. 16.9*a*). Gauges of this type are mainly employed for controlling the highest level of material in bins. When the bin is filled up to the float, this is deflected towards the wall and its mercury contact in a protective cap near the pivoted joint is closed, thus closing an electric circuit which sends a warning signal. In systems with complete automatic control, the electric circuit sends a command to the feeder which is either stopped or switched over to another bin cell.

Another level gauge of the mechanical type has a sensing membrane or blade which can be arranged at various levels in the bin (Fig. 16.9*b*). As the material in the bin reaches the level of the gauge, it deflects the membrane or blade and the latter closes the contact to send a signal or command to controls.

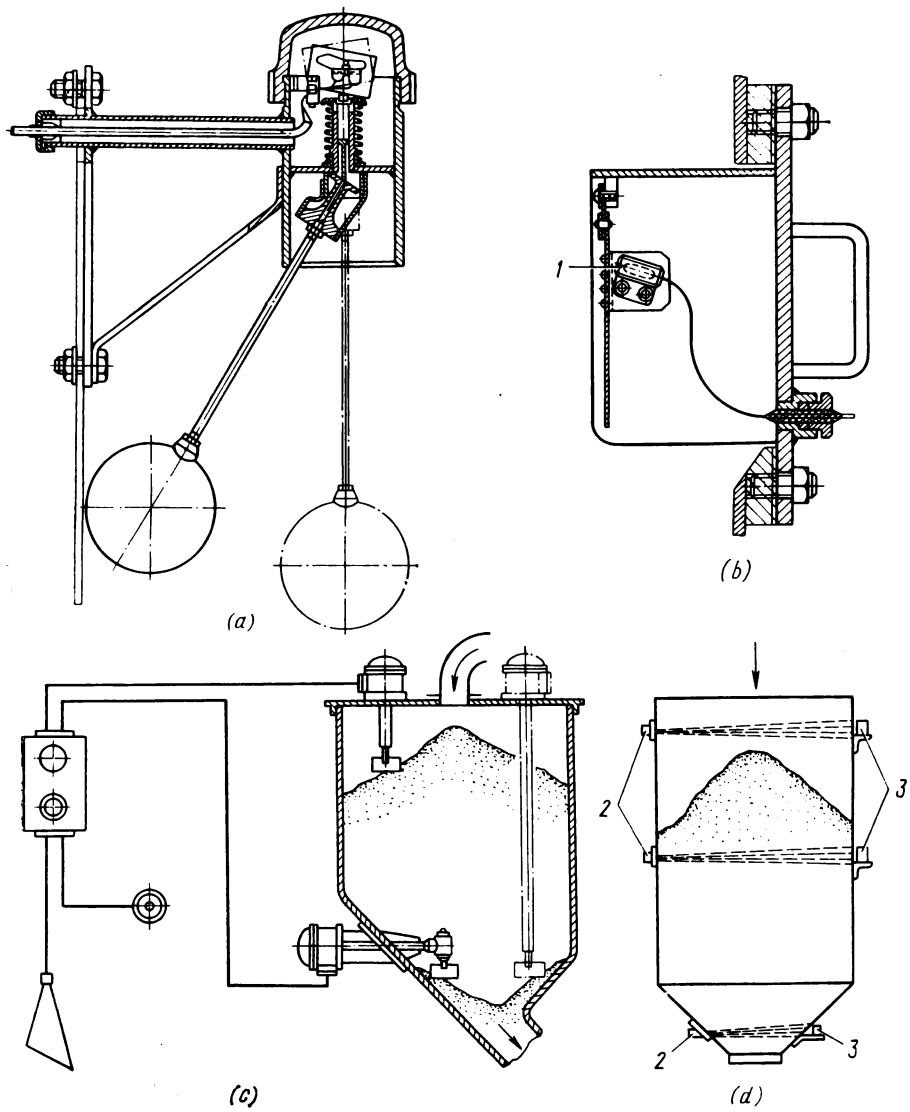
A third type of mechanical level gauge (Fig. 16.9*c*) has a bladed wheel mounted on a vertical shaft (or on a horizontal shaft passing through the side wall) and rotated by a small-power electric motor (up to 5 W). The motor stator is retained against rotation by a spring. When the level of the material reaches the wheel, this is stopped, which results in an increased electromagnetic interaction between the rotor and stator of the motor, so that the stator turns against the spring action through a certain angle and switches on a signalling device or the automatic control system of the conveyers which supply material into the bin.

Radioactive (gamma-emitters and gamma-indicators) and ultrasonic level gauges are more reliable in operation, since they require no mechanical contact with the material. Schemes of their arrangement for indicating the highest, medium and lowest level in a bin are shown in Fig. 16.9*d*.

With the application of level gauges, the operation of bins and the associated transport means can be fully controlled automatically. For instance, as a bin cell is filled, a command can be sent to a loading transport vehicle (or conveyer) to switch over to another cell; as a cell is emptied completely, the feeder under it is stopped automatically and the feeder of another (filled) cell is started, and so on.

16.3. Bin Gates

Bin gates open and close the discharge holes of bins and in some cases can



16.9. Level gauges for bins
(a) pendulum-type; (b) blade-type; (c) wheel-type; (d) gamma-indicator; 1—mercury contact; 2—gamma-meter; 3—gamma-indicator

trol the flow of a bulk material ›ugh holes. Accurate control of the ‹ rate by a gate is only possible ·n easily flowing bulk mate- s are used.

›in gates should satisfy the following ıirements: ease of operation, quick on, proper tightness, and the possi- ty for controlling the flow rate of the material. Besides, they should be simple in design, sufficiently strong and have small dimensions where possible. In some cases, they are required to be adaptable to remote and automatic control.

Bin gates can be operated manually or by electrical, pneumatic or hydraulic mechanisms. All types of gates with

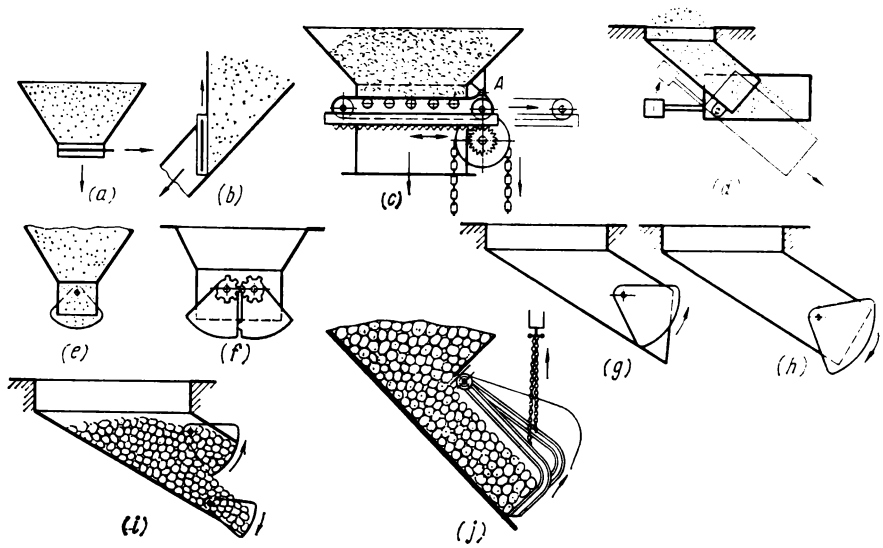


Fig. 16.10. Schemes of bin gates
(*a*), (*b*) slide gates; (c) caterpillar belt gate; (*d*) tray gate; (*e*) single-sector gate; (*f*) double-sector (jaw) gate; (*g*) and (*h*) inclined sector gates with upward and downward opening motion; (*i*) doubled sector gate; (*j*) finger gate

mechanical drives can be remote-controlled.

An electric drive for bin gates has an electric motor and reducer gear connected through a torque-limiting clutch to prevent occasional breakage.

Direct-action drives (pneumatic and hydraulic) are advantageous over the electric type, since they suit better to the pattern of motion of bin gates (quick starting, short fast travel, and suddenly increasing resistance on jamming of large lumps). Their working member is a pneumatic or respectively hydraulic cylinder with a piston. Pneumatic drives are especially convenient in pneumatic transport system where compressed air is readily available. A drawback of pneumatic drives is freezing of the moisture contained in compressed air which can occur in winter time in systems operating outdoors or in unheated premises. Under such conditions, hydraulic drives with oil as the working liquid turn out to be more reliable.

By the operating principle, bin gates can be divided into two main groups: those which cut off the flow of material and those which develop a back pressure when being closed. The first, more numerous, group includes various gates with a flat closing member and sector gates and the second group, tray gates.

Bin gates with flat members (Fig. 16.10*a* and *b*), which are mounted in the bottom or side walls of bins, are more compact, but have an essential drawback: high resistance in their slots during opening or closing; for that reason they are only employed in small-size discharge holes or with light pressures of the material on the gate.

A variety of a flat-member gate is the caterpillar belt gate (Fig. 16.10*c*) which comprises an endless conveyer belt fixed at one point *A* at the edge of a bin hole and a movable frame with two small-diameter pulleys and support rollers. As the frame is moved, the belt revolves on the pulleys and also moves and thus opens or closes the hole. The belt simply rolls on the rollers, but does not slide on the material.

Sector gates (Fig. 16.10*e-i*) have a cylindrical working surface which tilt-

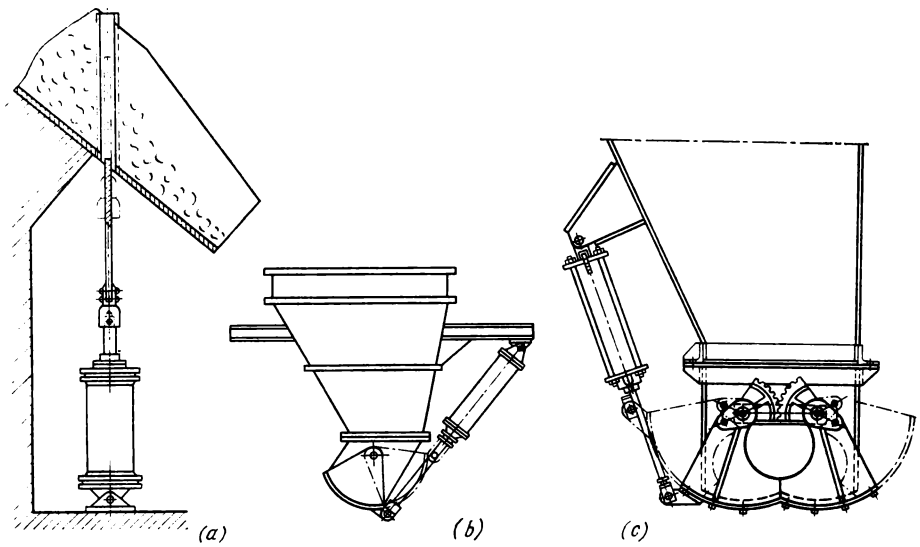


Fig. 16.11. Pneumatically driven bin gates
(*a*) slide gate; (*b*) single-sector gate; (c) jaw gate

on a horizontal shaft to close or open the bin hole; since sector gates have no slots, their resistance is much lower than that of flat-member gates. Sector gates may be with one or two sectors (jaw type) and with upward or downward opening motion of the sector (Fig. 16.10*g* and *h*).

A sector gate which opens upwards above the inclined plane on which the material slides makes it possible to control the flow rate of the material by closing the sector partially. Its drawback is that the sector can jam material lumps when being closed. A sector gate with downward opening motion of the sector is devoid of this drawback, but offers a high resistance to the material flow when opened partially and thus is poorly suitable for controlling the material flow rate.

Doubled sector gates (Fig. 16.10*i*) have the advantages of the both types mentioned above and are employed with lump materials. For opening, first the smaller (lower sector) is opened and the flow rate of the material is controlled by the position of the upper sector: to close the bin, the upper sector is first lowered (but not up to the slide plane), after which the lower sector is raised into the closing position and retains the material in the bin.

A variety of sector gates is the finger gate shown in Fig. 16.10*j*. It has a number of pivoted heavy levers (fingers) sharpened at the ends and suspended on chains. When being lowered, the fingers penetrate easily the layer of material and can press large lumps to the chute surface and thus close the bin hole. For opening, the fingers are raised by chains from a common drive: first those which have penetrated deeper into the material and then all together.

A bin gate of the second group, i.e. with the formation of a back pressure on closing, is illustrated in Fig. 16.10*d*. Gates of this type do not jam lumps and can control the flow rate of the material by changing the angle of incline of the tray. They have, however, too large overall dimensions.

The arrangements of various gates (flat-member, single-sector, and jaw-type) with pneumatic drives are illustrated in Fig. 16.11.

The pressure of bulk material on a gate can be determined in the same way as the pressure on the walls or bottom of a bin. If F is the cross-sectional area of the discharge hole, the force of pressure on the gate $P = Fp$, where p is the pressure of the bulk load.

The quantity of bulk material discharged through a gate is the product of the cross-sectional area of the hole by the speed of outflow of the material. In contrast to the speed of liquid flow, the speed of an outflowing bulk material is lower at a higher coefficient of internal friction of the material. This circumstance is considered by introducing a coefficient λ_{fl} into the formula for liquid flow (see in the course on hydraulics):

$$v = \lambda_{fl} \sqrt{2pg/\rho} \qquad (16.10)$$

According to some authors, λ_{fl} is equal to 0.65 for dry sand, 0.55 for grain, 0.4 for coal (with fines), 0.3 for ungraded ore, and 0.2 for wet sand and sawdust.

16.4. Feeders and Meters

Feeders are motor-driven mechanical devices which are mounted at the discharge holes of bins and funnels and ensure uniform and controlled delivery of bulk materials from them. In some cases, for instance, when reloading the material onto a conveyer, feeders impart to the material flow a required speed in proper direction. The throughput capacity of feeders is controlled by the gate or by varying the operating parameters of the feeder proper.

Most types of feeder can receive the material from bins without gates; on a stop, they retain the material by the back pressure they form. On the other hand, material discharge from bins by means of feeders is characterized by active influence of the feeder elements on the load, which is of special importance for poorly flowing materials which can hardly be discharged from bins solely by the force of gravity.

Feeders are diverse in design and are selected depending on the kind and properties of a particular material, the required throughput capacity, and local conditions.

A large group of feeders are varieties (counterparts) of conveyers: belt, apron, flight, screw, rocking, shaking (Fig. 16.12*a-f*). Feeders differ from the respective conveyers by a shorter length, elevated power of the drive, a higher strength (in order to withstand the pressure of the material under the bin gate), can carry materials in thicker beds, and overcome greater resistances in operation. The feeders of this group sometimes serve also as conveyers, i.e. they transfer the material to a certain distance from the bin hole.

Another group of feeders (drum-type, disc, chain Fig. 16.12*g-i*) has no counterparts among conveyers and can only deliver the material directly at the bin hole.

Belt feeders (Fig. 16.12*a*) may be horizontal, inclined or declined. As compared with belt conveyers, they have more closely spaced roller supports of the carrying run and no supports under the return run (if they are not too long). Belt feeders are usually provided with stationary skirtboards and mostly (except for accelerating feeders) have a low belt speed (0.1-1 m/s). They are mainly employed for the transfer of granular and small- and medium-size lumpy materials. The throughput capacity is controlled by a cut-off plate or by varying the belt speed. Some belt feeders are equipped with automatic scales whose indications are used to control the throughput capacity by varying the belt speed. Among the advantages of belt feeders are a high reliability, small mass and a wide range of throughput capacities.

Apron feeders (Fig. 16.12*c*), like the belt type, can be horizontal or incli-

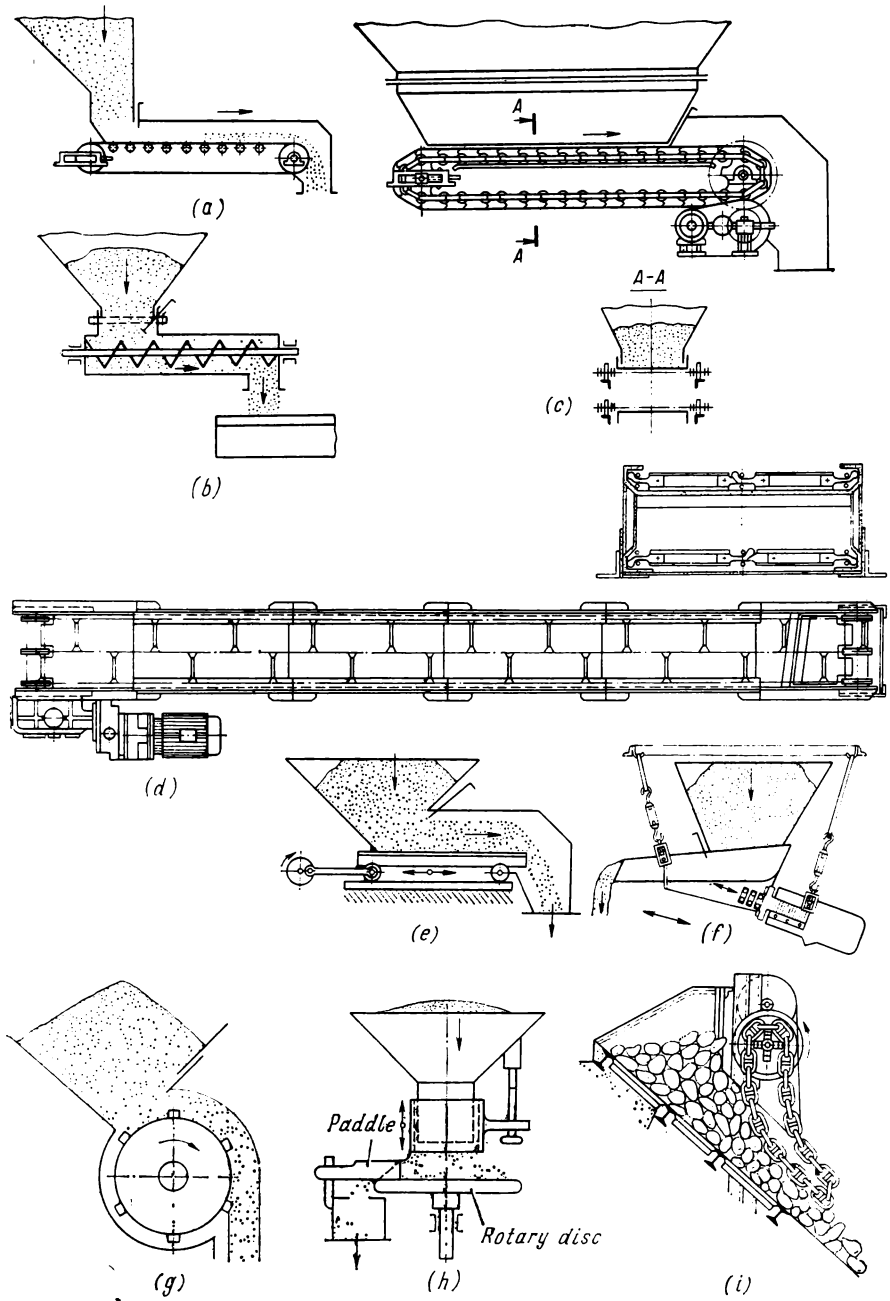


Fig. 16.12. Schemes of feeders
(*a*) belt type; (*b*) screw; (c) apron; (*d*) flight; (*e*) rocking; (*f*) shaking; (*g*) drum type; (*h*) disc type; (*i*) chain type

ned, but allow a higher angle of incline. They may be on moving or stationary roller supports, with side walls or stationary skirtboards. The apron speed is usually 0.05-0.25 m/s.

Apron feeders are characterized by a high strength and can be used for carrying heavy, large-lump and abra-

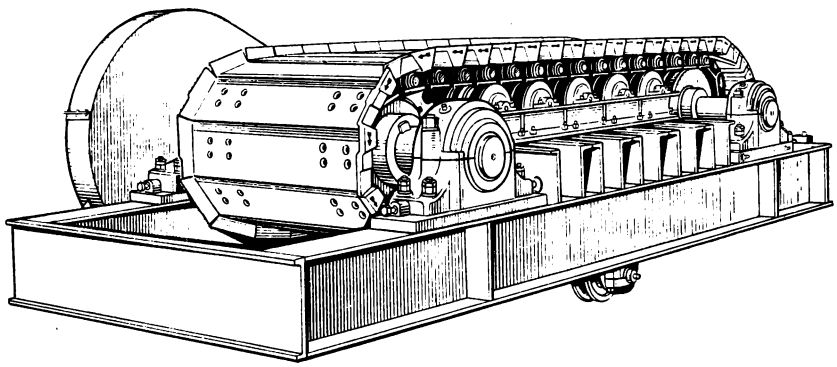


Fig. 16.13. Heavy-duty apron feeder

sive loads, such as ore, crushed stone, etc. With poorly flowing materials, they can replace the bin bottom (as shown in Fig. 16.12*c*) and absorb impacts of lumps and the pressure of the load column. Their drawbacks are a large mass and high cost.

A heavy-duty apron feeder is shown in Fig. 16.13. The carrying run of the apron moves on a set of stationary rollers mounted on transverse beams of the frame.

Flight feeders (Fig. 16.12*d*) have, depending on the required width, three to six hauling round-link chains and two-five staggered rows of flights. Like apron feeders, they are applicable under the heaviest operating conditions, but have a simpler design and lower height. Their drawbacks include rather quick wear of the supporting plates and the probability of jamming of material lumps between the chains and sprockets. The throughput capacity and the required power of drive motors of belt, apron and flight feeders are determined as for similar conveyers, with due regard of their specific operating conditions.

Screw feeders (Fig. 16.12*b*) have a continuous solid screw mounted on two bearings and rotating in a closed trough or pipe. The screw has a short lead to facilitate motion of the material or sometimes is made with two leads. The filling coefficient of the trough (pipe) is rather high: $\psi = 0.8\text{-}0.9$. Screw feeders are employed for carrying pulverized, granular and fine-size lumpy bulk loads.

Rocking feeders (Fig. 16.12*e*) have a horizontal or declined trough with side walls or skirtboards, which is supported by stationary rollers or wheels or suspended on ties and can be reciprocated by a crank mechanism. In a straight motion of the trough, the material on it is involved into motion due to friction. As the trough moves forward, a portion of material is fed into the freed space on it from a bin. In a return stroke, the material cannot move backward owing to the back pressure created by the rear wall of the bin funnel, and therefore, is poured partially over the front edge of the trough. If V' (m^3) is the volume of the material poured from the trough in each stroke (which is proportional to the layer height, trough width, and the length of stroke), and n is the number of strokes per minute (usually $n = 20\text{-}60$), the throughput capacity (m^3/h) of the feeder

$$V = 60V'n \qquad (16.11)$$

The throughput capacity of rocking feeders can be varied by a cut-off plate or by changing the trough stroke. Rocking feeders are employed with graded and ungraded bulk loads with small, medium or large lumps. They have a wide range of throughput capacities, but are inapplicable with sticky

loads. Their advantage is a simple and strong design.

Shaking feeders (Fig. 16.12*f*) operate by the same principle as shaking conveyers and have essentially the same centrifugal (unbalanced) or electromagnetic vibration exciters, supports, and elastic elements. Their throughput capacity is usually a few tens t/h and, in the heaviest designs, up to 2000 t/h. Shaking feeders can simultaneously serve as screens, for instance, to screen through fines before feeding the material onto a belt conveyer. An advantage of shaking feeders, as of the rocking type, is that the discharging edge of the trough can be arranged quite low above the receiving belt conveyer, and therefore, the belt will not be damaged by falling lumps.

The throughput of shaking conveyers is controlled automatically by changing the amplitude and frequency of trough vibrations.

Drum feeders (Fig. 16.12*g*) are quite simple in design. The drum has a smooth surface (for easily flowing granular and small-size lumpy materials) or ribbed (for large-lump loads). The throughput capacity of a drum feeder is proportional to the height of the load layer and the circumferential speed of the drum.

Rotary-table, or disc, feeders (Fig. 16.12*h*) have a round table (disc) rotating on a vertical axis just under a cylindrical telescopic pipe connected to the round discharge hole of the bin. As the table rotates, it receives the material from the bin, while an oblique plate mounted at its side scrapes off part of the material from the rotating table. The throughput capacity is varied by rearranging the telescopic pipe or plate. Feeders of this type are applicable with easily flowing pulverized, granular and small-size lumpy materials.

The rotational speed of the table is chosen so that material particles could not be thrown off from the table by centrifugal forces, i.e. with particles of a mass m, table radius r, rotational frequency of the table n, and the coefficient of friction of particles on the table f, the following condition must be observed:

$$mgf > mr\,(2\pi n)^2$$

whence

$$n < \frac{1}{2\pi}\sqrt{\frac{gf}{r}} \qquad (16.12)$$

Chain feeders (Fig. 16.12*i*) have a number of heavy chain sections fastened at one end to a power-driven drum and connected with one another by transverse round links, thus forming a 'curtain' which covers the discharge hole of a bin and rests by the end portion on the layer of material in a chute, preventing the material to flow by gravity. As the drum rotates, the chains press the layers of the material to the chute and control the speed of material sliding. The mass and length of chain sections are selected depending on the size and mass of load lumps.

Chain feeders are used with large-lump materials of more or less uniform size of lumps. Fine particles, if present in the material, can pass freely through the curtain, while individual very large lumps can raise the curtain and move at a higher speed. For a given mass of chains, the throughput capacity of a chain feeder depends on the circumferential speed of the drum.

A special group is formed by unloading feeders for slotted bins (see Fig. 16.5*b*). A feeder of this type moves along the shelf under the slot of a bin and grabs the material by paddles or pushes it through onto the opposite side. Another type of bin unloader is shown in Fig. 16.14. It has a power-driven undercarriage with two discs rotating on inclined axes. The discs enter the slot from the rear side of the bin. During motion of the undercarriage, the discs roll over the material and throw it from the shelf onto a parallel conveyer. Unloaders of this type can operate with poorly flowing and large-lump materials.

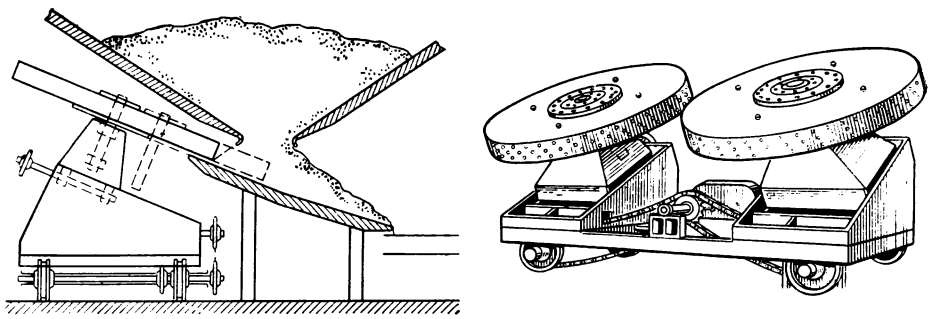


Fig. 16.14. Disc-type unloader for slotted bins

In many cases, feeders are required which can maintain automatically the desired throughput capacity. By the principle of action, they are divided into two types: (a) direct-action, with a weighing element which is connected by a kinematic linkage with a member to control the material flow (Fig. 16.15*a*) and (*b*) indirect-action, which also have a weighing element, but here the controlling member is actuated from an individual power source (see Fig. 16.15*b*).

The feeder of the first type is mounted on one arm of a rocker frame, the other arm carrying a counterweight. The frame is connected to the level mechanism of the bin gate, so that the gate opening can be varied by moving the counterweight on the frame. By moving the counterweight farther from the frame centre, the thickness and mass of the material layer on the belt increase to balance the system, resulting in a higher throughput capacity of the feeder.

In the second type of feeder (Fig. 16.15*b*), the frame of a short weighing belt conveyer is hingedly fastened at one end and is supported by a magnetoelastic weight sensor at the other. The material is supplied onto the weighing conveyer by a controllable electromagnetic shaking feeder. The sensor sends a control pulse into a control device which can vary the current fed to the electromagnets of the shaking feeder and thus change the feeder throughput capacity.

Meters are cyclically operating devices which remove in every cycle a measured quantity (portion) of bulk material from beneath a bin. The mate-

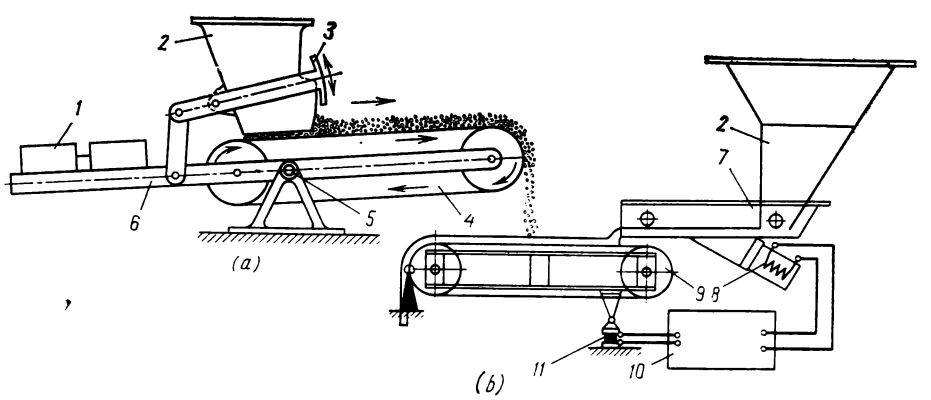


Fig. 16.15. Feeder with automatically controlled material flow

(*a*) direct-action feeder; (*b*) indirect-action feeder; *1*—counterweight; *2*—bin; *3*—gate; *4*—belt; *5*—hinged support; *6*—frame; *7*—shaking feeder; *8*—electromagnetic vibrator; *9*—weighing belt conveyer; *10*—regulator; *11*—magneto-elastic sensor

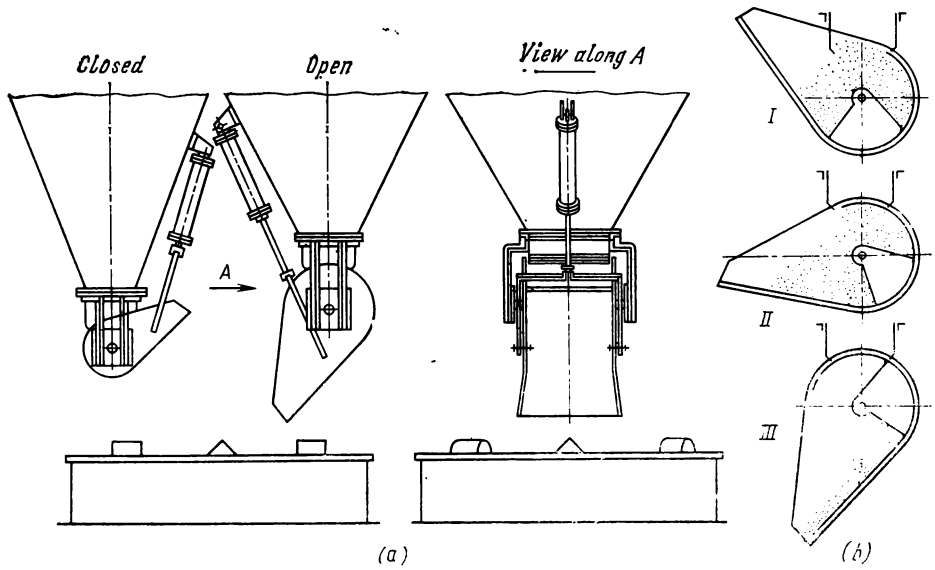


Fig. 16.16. Volume meter
(*a*) general view; (*b*) positions of tilting container (*I*—charging, *II*—tilting, *III*—discharging)

rial can be metered by mass or by volume. Volume metering is done by means of a vessel (container) of a definite volume (Fig. 16.16), a plunger or a sector drum with cells. The container can be turned by a pneumatic cylinder so that its cylindrical rear wall closes the bin hole.

Mass metering is effected by means of a container mounted on scales. The scales actuate the bin gate or feeder at the moment when the mass of bulk material in the container reaches the given value. The discharge hole of the container is opened and closed automatically.

Automatic mass metering can also be done by means of conveyer scales which automatically stop a conveyer or feeder upon passing a measured mass of load.

16.5. Throwing Machines

In a throwing machine, a bulk load is accelerated and acquires a kinetic energy sufficient for its free directed flight to a required distance (usually within 20-30 m). Throwing machines are employed in various branches of industry; they are especially efficient when bulk materials must be delivered to poorly accessible places. In mining, throwing machines are used for filling in worked-out faces (stowing machines) and in open quarries, for spoil heap filling. In metallurgy, they are used for charging the furnaces. In road construction and earth moving, earth throwers build up embankments, road beds, etc. Throwing machines are also used as loaders in railway and water transport, as piling machines in stores of coal and other bulk materials, as grain throwers in granaries, etc.

By the operating principle, it is distinguished between throwing machines in which the material is accelerated due to friction on the working member, those in which it is accelerated by gripping and pushing, and those of combined type in which both principles are utilized. By the kind of working member, continuous-action throwing machines are divided into three types: belt, rotor-blade, and disc type.

Belt-type throwing machines may be with a straight or bent belt. A straight-

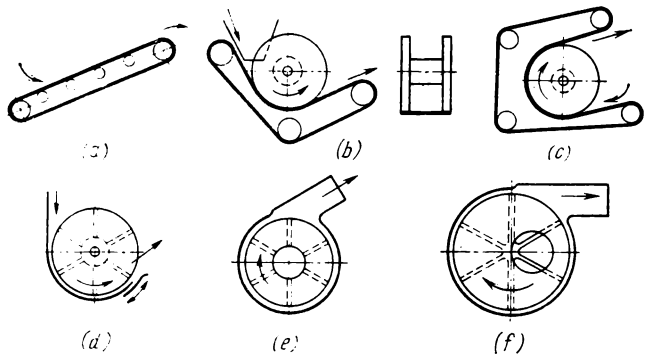


Fig. 16.17. Schemes of throwing machines
(*a*) belt type with straight belt; (*b*) and (*c*) belt-and-drum type; (*d*) and (*e*) rotary type; (*f*) disc type

belt thrower (Fig. 16.17*a*) is essentially an inclined belt conveyer with the belt moving at a high speed, usually 15-20 m/s. To accelerate a bulk load to such a high speed, the length of the machine should be quite large, which makes it poorly transportable. The length of belt throwers is sometimes decreased by providing cleats on the belt which, however, spread the material in a wider cone.

Throwing machines with a bent belt, or belt-drum machines (Fig. 16.17*b* and *c*), are more compact, since the force of friction of the load on the belt is increased in them substantially owing to the effect of centrifugal forces. The belt bends around disc-shaped flanges of a drum and the material is poured into the gap between the belt and the central portion of the drum. The material can be ejected either from beneath the drum (Fig. 16.17*b*) or above it (Fig. 16.17*c*). The latter type has an advantage that the belt can be wrapped around the drum over a greater angle, but involves certain complications in material feeding to the drum (the material must be given from the same side to which it is thrown). In throwing machines with the material ejected from beneath the drum, the material can slide on the wall of the feeding bin and acquire an appreciable speed before entering the curved portion of the belt, where its speed increases further and the material is thrown along an arc in the desired direction.

Two schemes of rotor-blade throwing machines are shown in Fig. 16.17*d* and *e*. In the first machine, the bladed rotor is mounted on a horizontal shaft and the material is fed tangentially to its periphery and can be thrown with a high initial speed. A drawback of throwing machines of this type is that lumps can be jammed between the blades and rotor housing. In the second machine, the rotor may be arranged either horizontally or vertically and the material is introduced at a low speed through a central hole in the housing and is then accelerated when moving radially along the blades from the centre to the periphery. As in belt-type machines, the jet direction in rotor-blade machines can be controlled (by means of a cut-off plate or by turning the entire housing of a machine).

A third type of continuous-action throwing machines, the disc-type machine (Fig. 16.17*f*), differs from the horizontal rotor-blade machine in that instead of a stationary circular bottom it has a flat or cone-shaped rotating disc with a smooth surface or radial blades which grab the load and throw it through a discharge hole in the cylindrical wall of the housing; the material is ejected radially in all directions (such machines are used, for instance, for loading cargo holds on a ship from central points).

Four particular embodiments of throwing machines are illustrated in

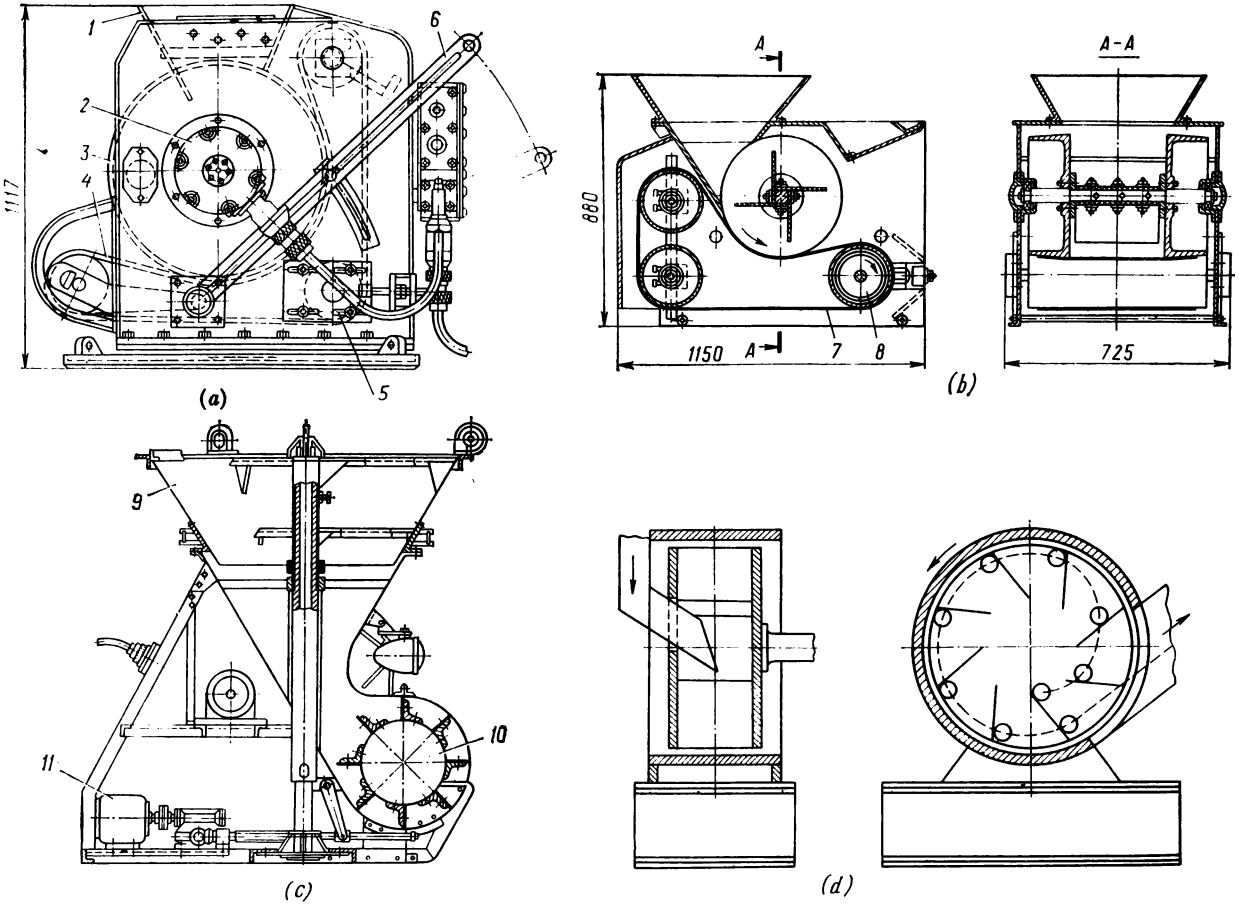


Fig. 16.18. Types of throwing machines
(*a*), (*b*) belt-and-drum stowing machines; (c) rotary-type ship-loading machines; (*d*) rotary-type for fettling of open-hearth furnaces

Fig. 16.18. The belt-drum stowing machine (Fig. 16.18*a*) can be moved on its supporting platform on the ground surface as the filling operation proceeds. It has two disc-shaped pulleys *3* mounted on a single shaft with a drum and rotated from individual electric motors through a built-in gear transmission.

The material is fed through a funnel *1* and catched by moving belt *4*. The outlet angle can be controlled by lever *6* which changes the position of the head drum *2*. The machine has also a take-up pulley *5*. The throughput capacity of the machine is 80 m^3/h with the belt of a width of 325 mm and belt speed 15 m/s. Each of the motors has a power of 5.5 kW. The size of lumps of the material should not exceed 100 mm.

The belt-drum machine shown in Fig. 16.18*b* has an additional bladed shaft. The belt *7* is driven by an electric motor-pulley *8*. The main characteristics of the machine are as follows: belt width 500 mm, belt speed 17 m/s, power of motor-pulley 8 kW, mass of machine 95 kg, and the largest size of lumps 100 mm.

A high-capacity blade-type throwing machine for filling in ship holds (Fig. 16.18*c*) is mounted on a vertical shaft and can be turned on it together with a feeding funnel *9*. The bladed rotor *10* has interchangeable rubber-faced blades. The throwing angle can be changed by a cut-off plate driven from a separate electric motor *11*. The throughput capacity of the machine (for heavy grain) is 200 t/h. The machine is remote-controlled.

Another throwing machine (Fig. 16.18*d*) differs from the previous types by charging the material through a central hole. It is employed for fettling of open-hearth furnaces. A typical feature of the machine is that the blades are inclined at 12-15° from the radial directions. The dotted line in the figure shows the path of a particle in the machine. The machine ejects the material at a speed of 13.5 m/s and has a throughput capacity of 36-42 t/h.

A common drawback of all types of throwing machines is quick wear of the elements in contact with the material, especially when handling lumpy and abrasive loads. The belts of machines operate under very heavy conditions, since they come in contact with the material and bend at high speeds around small-diameter pulleys. For that reason they should contain not more than three plies and be protected on the working surface by thicker covers. Rotor blades and the internal surface of housings of blade-rotor machines should be lined with wear-resistant materials. In many cases, the crushing effect on the materials due to impacts and rubbing should also be considered. In that respect, belt-type throwing machines are preferable over the rotor-blade type.

16.6. Automatic Scales

Automatic weighing of a transported bulk material can be carried out directly on a conveyer or on a separate weighing device.

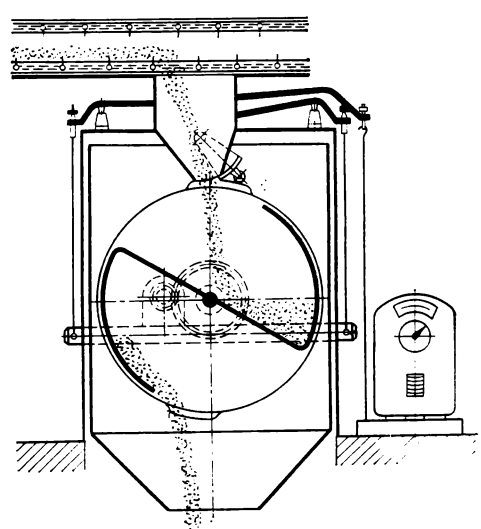


Fig. 16.19. Periodic-action scaling device (with rotating drum)

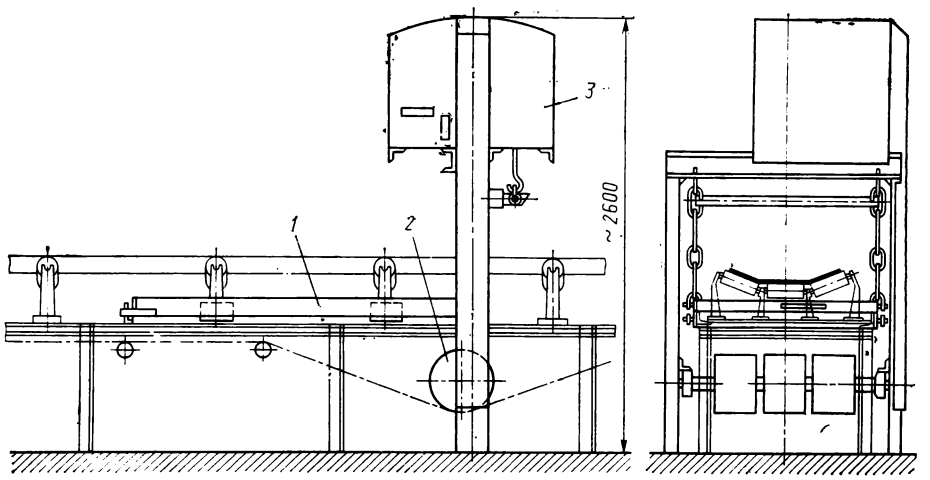


Fig. 16.20. Continuous-action conveyer scales

By the operating mode, automatic scales are divided into continuous and periodic, and by the method of weighing and type, into mechanical and electronic (electrical). Besides, radioactive and ultrasonic meters come into use, which measure the volume of bulk materials transported on conveyers.

Periodic-action mechanical scales in the form of a turning drum with two sections are shown in Fig. 16.19; they are applicable with dry easily flowing bulk materials. The material is delivered by an immersed-flight conveyer and discharged into an intermediate reservoir with a sector gate. As the drum is turned clockwise, a lug on its periphery opens the gate and the material flows into the drum section that is in the upper position at that moment. As the drum turns further through a certain angle, the feeding gate closes and cuts off the flow of material, while the portion of material contained in the opposite drum section is discharged. At that moment, the weight of the material portion in the upper drum section is measured by the scaling mechanism from which the drum is suspended, and the scale indications are automatically summed by a counter-type adder.

Bulk loads can be weighed directly on a conveyer by means of conveyer scales (Fig. 16.20) which can be mounted on a horizontal or inclined conveyer section. The scales have three main units: (1) a platform *1* supported by prisms at one end and suspended on ties from scales on the other end; the platform carries two roller supports for the carrying run of the conveyer belt; (2) a scaling and recording device *3* contained in a closed box and mounted on a stand; and (3) a drive mechanism *2* with a pulley rotated by the return run of conveyer belt. The scaling device measures the mass of the load on the conveyer belt every time when the belt moves through a distance equal to the spacing between the two roller supports on the platform, while a counter, which is linked kinematically with the pulley of the drive mechanism, records the scale indications and sums the products of the distributed load on the belt by the belt speed, i.e. determines the mass throughput capacity.

The conveyer scales have an accuracy of $\pm 1\%$ for loads ranging from 50% to 100% of conveyer capacity. They are, however, rather intricate in design and have large dimensions (length and height).

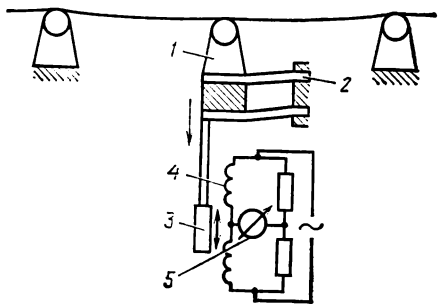


Fig. 16.21. Scheme of electronic weighing device
1—weighing roller support; *2*—leaf springs; *3*—steel core; *4*—windings; *5*—indicator

In recent time. more simple and compact electronic scales find ever wider application. They can perform instantaneous weighing of loads on a conveyer and integrate the mass indications in time. The pressure of a distributed load on conveyer belt is transmitted to a weighing roller support and from this to supporting elements which are thus subject to elastic deformations. The stresses and deformations in the elements are measured by a sensor which sends corresponding pulses to an electric circuit. An indicator gives the mass of load per metre of the conveyer belt at the moment of measurement. At the same time, a pulley or roller, which is pressed to the belt and rotates without slipping, measures the belt speed, and therefore, the mass throughput capacity of the conveyer.

Electronic scales employ three types of sensors (transducers): inductive, magnetoelastic, and strain-gauge type.

An inductive sensor (Fig. 16.21) has two windings of an electric circuit and a steel core connected to a deformable support element. The latter deforms under the gravity force of the conveyer load and moves the core relative to the windings, thus changing the inductance of windings and the current in the circuit, which is measured by an indicator.

Magnetoelastic sensors are made of ferromagnetic materials. Their operating principle is based on that, as a tensile or compressive stress appears in them, it changes the magnetic permeability, which causes corresponding changes of the resistance in the electric circuit.

Strain gauges, which are used most widely in conveyer scales, are essentially loop-shaped elements made from very thin wire. A strain gauge is glued onto the surface of an element which can deform under a load above it. In deformation, straight portions of strain gauge wires are tensioned or compressed, thus changing the total resistance in the electric circuit. An amplifier is usually connected into the circuit to increase its sensitivity.

The volumes of non-magnetic bulk materials carried on a conveyer can be measured by means of electronic gamma-indicators (Fig. 16.22*a*). A radioactive source *1* directs gamma-rays to

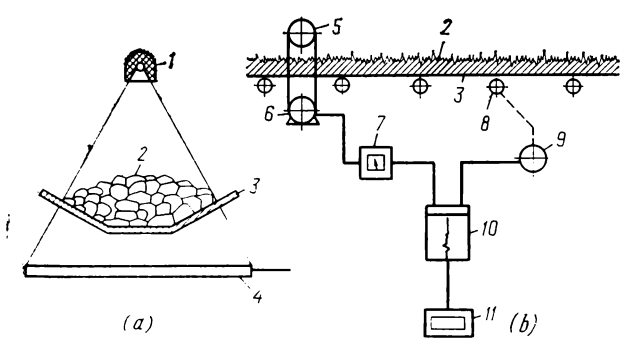


Fig. 16.22. Scheme of a radioactive device for measuring the throughput capacity of conveyer

the conveyer belt *3* with bulk load *2* on it. A radiation detector *4* is arranged under the belt. The electric signal formed in the detector is a function of the quantity of material on the belt; if the belt speed is measured simultaneously, it is possible to determine the throughput capacity of the conveyer.

The scheme of a radioactive arrangement for measuring the throughput capacity of a conveyer is shown in Fig. 16.22*b*. It has two transverse horizontal tubes *5* and *6* mounted above and below the conveyer belt *3* with load *2*. One of the tubes contains a radioactive emitter and the other, a radiation detector which transforms radiation pulses into electric signals. The intensity of the radiation flux received by the detector depends directly on the thickness of the material layer on the belt.

The detector sends electric pulses through amplifier *7* to a recorder *10* which receives another pulse from a tachometer *9* which measures the rotational frequency of a roller *8* pressed to the belt. A counter *11* records the total quantity of the material carried by the conveyer.

The conveyer scales determine the throughput capacity of a conveyer at the point of their installation and the quantity of material carried in a definite time interval. Their indications can also be used for automatic control of conveyer productivity, for instance, by varying the speed of the conveyer motor.

16.7. Continuous-action Loading Machines

Loading machines (loaders) are used for mechanized handling of bulk materials in stores, construction sites, loading stations in railway and other means of transport, for handling of ores in mines and quarries, etc. In addition to mobile conveyers which must be loaded by special mechanisms, loading machines can themselves take the material from piles by their working (grabbing or scooping) members. Loading machines are usually made self-propelled to increase their manoeuvrability. Their principal parts are, in the general case, a working (grabbing or scooping) member, conveying transfer element, and a self-propelled undercarriage. In some types of loading machines, a single member can take the material from a pile and carry it to the unloading place.

By the operating mode, loading machines can be divided into continuously and periodically acting. In periodic-action machines, the working member is made in the form of a bucket, scraper or rake. In continuous loading machines, working members are more diverse in design (see later in this section).

By the kind of power supply, loading machines are divided into those supplied from an external power source (the main type) and those with an autonomous (independent) power source. The former include electrically driven machines and the latter are usually driven by internal combustion engines. Machines of the latter type are employed rarely, only in cases when electric power is not available or the machine has to operate over a wide front and an electric cable would restrain its freedom.

The main units of a machine, i.e. the working members, conveying means (one or two conveyers), and the truck, can be driven by a common (group) drive or individually. In modern machines, individual drives are preferred, though they have certain drawbacks, in particular, a higher installed power and more intricate electric circuitry.

Some novel designs of loading machines have an electrohydraulic drive with an electric-motor driven high-pressure pump supplying the fluid to hydraulic motors and cylinders of machine members. An advantage of the electrohydraulic drive is that hydraulic cylinders can be easily connected to machine elements (working member,

conveyers, etc.) to cause them to perform various auxiliary movements (lifting, turning, etc.).

The working (pickup) members of continuous-action unloading machines can be divided into three groups by the method in which they pick up the material: those which dig in under a pile; those which take the material from the top; and those which gather the material from the sides to centre. The principal types of continuous-acti-

Table 16.1. Principal Types of Continuous-action Pickup Members

Method of taking the material	Pickup member	
	type	scheme in Fig. 16.23
Digging from below	Multi- bucket chain or belt	*a*
	Shaking or rocking nose	*b*
	Multi-bucket rotor	*c*
	Toothed rotor	*d*
Scooping from the top	Flight rotor	*e*
Gathering from sides	With two gathering arms	*f*
	Cantilever scrapers on two endless chains (jib-type)	*g*
	With one or two discs	*h*

on pickup members are indicate Table 16.1 and shown in Fig. 16.2

In some cases, the main pi member is supplemented with au ary members (for instance, scoo screws in a multi-bucket loade a shaking nose in a gathering-arm chine).

The loading (pickup) member is principal element of loading mach and determines their types. The of pickup member is selected depen on the properties of the transpo material (lump size, abrasiveness, and the required throughput capa of machine, though the selection these characteristics is not defi Only in rare cases the kind of dictates the particular type of pi member. For instance, multi-bu belt or chain machines are ma employed for handling easily flo bulk materials with small size of lu when the required throughput capa is not high. Toothed-rotor and scoop rotor machines can work with l lump materials. In many cases, l ever, the field of application of a pi member and the throughput capacit a machine depend not only on type, but also on the design of pickup member and the size of mach

Transfer conveyers in loading ma nes are usually of the belt, fligh apron type. A machine usually has or two and sometimes three success ly arranged conveyers. In mach

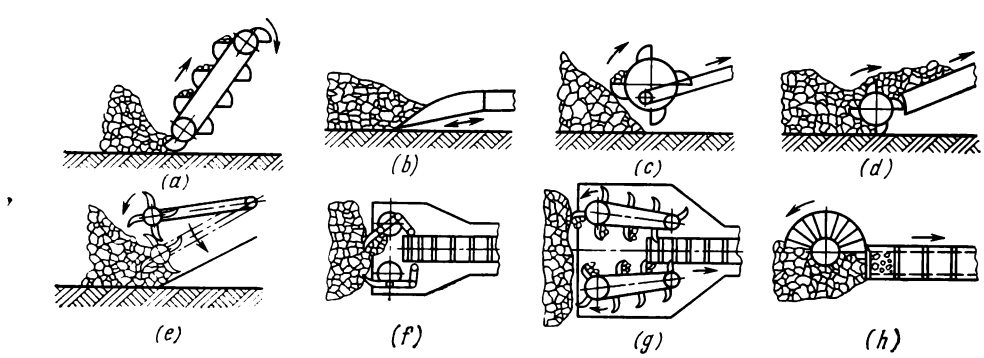


Fig. 16.23. Schemes of working members of loading machines

with two conveyers, the first (receiving) conveyer is usually associated directly with the pickup member; the second (unloading) conveyer is mounted on a cantilever arm and is usually tilting relative to the first vertically and horizontally. In some cases, the first (receiving) conveyer is also tilting in one or two planes in order to increase the loading front of a standing machine.

Transfer conveyers of a short length should accordingly have a low height, because of which their pulleys or sprockets have a diameter as small as possible. Cantilevered conveyers are usually driven at the tail end; a head-end drive is sometimes employed in belt conveyers with motor-pulleys. The belts and aprons of receiving conveyers are provided with low cleats to increase the angle of incline.

The undercarriage of loading machines may run on trucks (crawler-type), pneumatic tyres or railway wheels.

With a crawler-type undercarriage, the mass of a machine is distributed over a large area, which is advantageous if the machine has to operate on rough terrain. Owing to a high grip of the trucks, the machine can develop a high pressure on a pile and can move on steep slopes. The coefficient of gripping of trucks can be taken equal to 0.4-0.5 on sand, 0.5-0.6 on hard rocks, and up to 1 on clayey dirt road. However, crawler-type undercarriages are rather complicated and have a large mass.

Machines with pneumatic-tyre undercarriage can move quickly from one working place to another, i.e. have a high manoeuvrability (though somewhat lower than that of crawler-type machines). Pneumatic tyres have a high gripping coefficient which may vary within certain limits, depending on the kind of soil and the pressure in tyres. For instance, the gripping coefficient of low-pressure tyres (0.15-0.5 MPa) on compacted or dense soil is equal to 0.87-0.90 and that of high-pressure tyres (0.5-0.7 MPa) under the same conditions, 0.75-0.80. The gripping coefficient decreases if a layer of loose fines is present on the surface.

Loading machines on railway wheels are employed in cases where the materials are charged into railway carriages moving on the same track. Machines of this type are simple in design and can move quickly from one working place to another. Among their drawbacks are a limited loading front and a relatively low coefficient of gripping on rails, so that the mass of a machine is sometimes increased in order to develop a sufficient pressure of the pickup member. The gripping coefficient of wheels can be taken 0.25 on dry rails and 0.18 on wet rails. With a film of fine coal or quartz sand on the rails, the gripping coefficient increases respectively to 0.27 or 0.31 and 0.46-0.47.

Some loading machines will be briefly described below. The machine shown in Fig. 16.24 has the pickup member in the form of a multi-bucket chain elevator. The material is scooped by the buckets, lifted to a definite height and discharged onto a cantilever belt conveyer. The bucket elevator and the belt conveyer can tilt on the central axis of the discharging funnel to increase the operation front.

Loading machines of this type are employed with non-slumping bulk materials, such as sand, gravel, crushed stone, peat or coal. Their throughput capacity is around 50 m^3/h.

A loading machine type ПНБ-3М with two gathering arms is illustrated in Fig. 16.25*a*. It is intended for large-size lumpy loads, such as exploded rock, which are difficult to scoop by buckets. The machine is mounted on a crawler-type undercarriage. The two conveyers of the machine are of the apron type. The machine has an electric drive with the total power of electric motors 108 kW and an electrohydraulic drive of the tracks. Its throughput capacity is 3 m^3/min for iron ore or 12 t/min for coal. The machine mass is 20 t. Another version of this machine, type

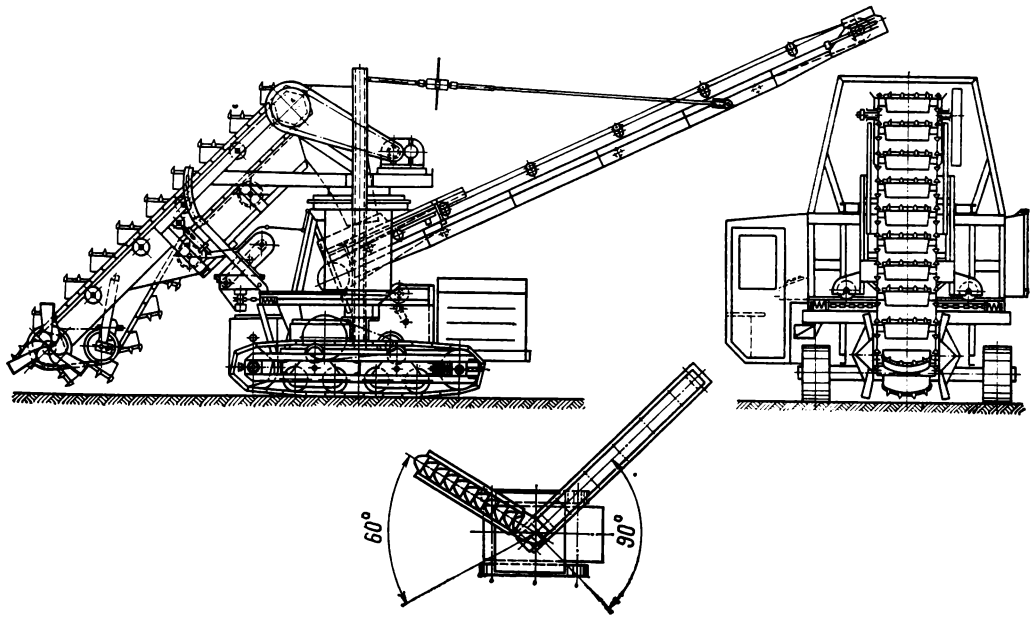


Fig. 16.24. Multi-bucket loader

ПНБ-4 has the following characteristics: total installed power of electric motors 240 kW, throughput capacity for ore and rock 6 m^3/min, mass 34 t, and the largest size of lumps 800 mm.

In some gathering-arm machines, each arm is essentially a link gear or four-link mechanism. For efficient operation of arms, the shape of their links and the kinematic of their motion are of high importance. The trajectory of the arm leading edges should be such that the arm first digs into the pile, scrapes the material along the inclined plate to the central axis of the machine, scoops it rearward onto the receiving conveyer and then returns into the initial position without entraining the material. The paths of the leading edges of gathering arms to satisfy these conditions are shown in Fig. 16.25*b*.

In bulk load stores, especially in coal stores, loading machines with a digging or scooping bucket-rotor member find ever increasing application. They have a wide spectrum of throughput capacities, from a few tens to a few thousands of cubic metres per hour. Two such machines: a small digging-loading machine and a powerful high-capacity machine for mechanized handling of coal in stores, are illustrated in Fig. 16.26*a* and *b*. Longitudinal coal piles in a coal store may be arranged at one or both sides of a stationary belt conveyer which is unloaded by means of a double-pulley unloading carriage. A gantry moves above the conveyer belt and carries a fully turnable and lifting cantilever beam with a belt conveyer and a bucket rotor at its end, which is balanced by a counter-weight. When delivered to the store, the coal from the stationary belt conveyer is discharged onto the belt of the machine and distributed between piles (the bucket rotor is then inoperative). When coal should be taken from the store, it is charged by the bucket rotor and the belt conveyer of the machine (which is then reversed) onto the stationary belt conveyer. This equipment increases substantially the capacity of a store, has a high throughput capacity,

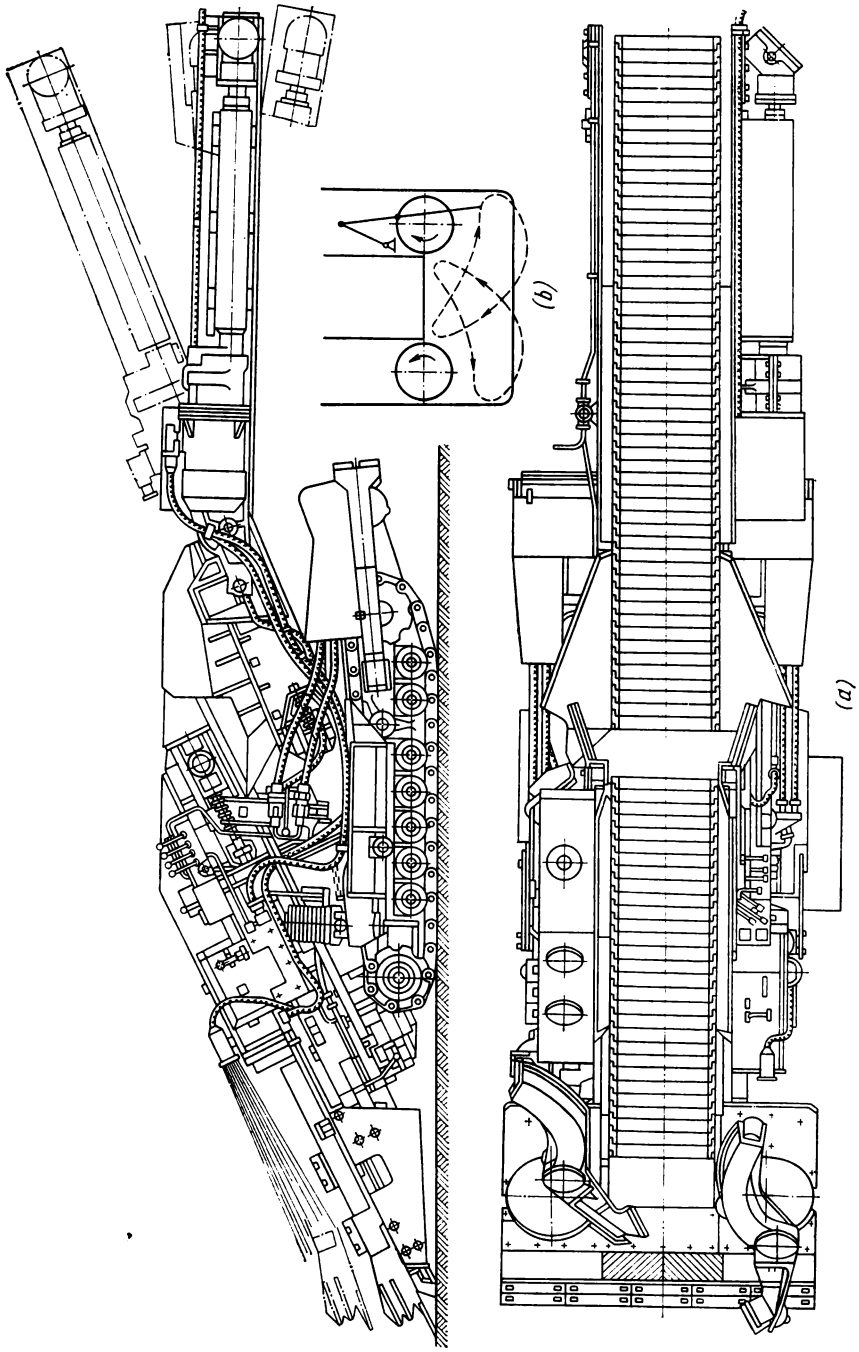


Fig. 16.25. Gathering-arm loader

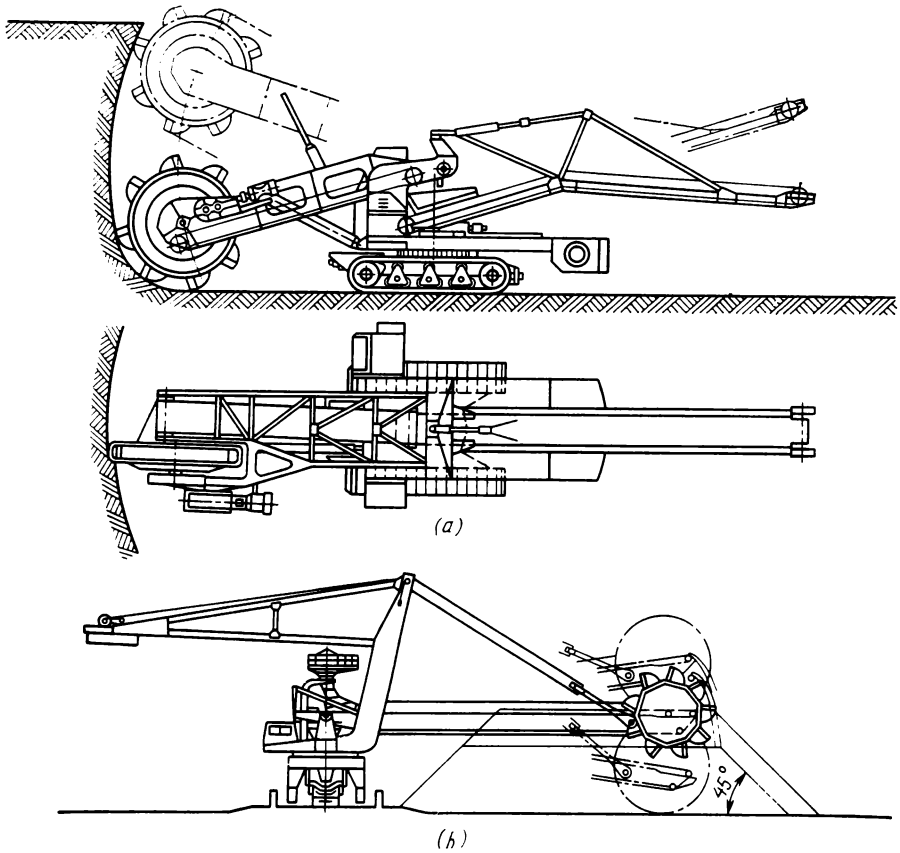


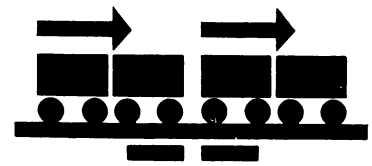
Fig. 16.26. Rotary loaders

and ensures complete mechanization of handling of bulk materials delivered to and from a store.

Modern loading machines should satisfy the following requirements: high reliability of operation with all possible states of the bulk materials (presence of large lumps, frozen or slumped lumps, etc.), convenient maintenance, good visibility and convenience for the operators, high repairability and interchangeability of main units and mechanisms, sufficient mobility in restricted conditions of stores, and minimum servicing personnel.

17

Conveyer Trains



17.1. Design

A conveyer train is a mobile (wheeled) section *1* (or a number of sections) which is moved on a rail track *2* by means of stationary drives *3* arranged along the path of motion of the section (Fig. 17.1*a*). The spacing between the drives *l* is smaller than the length of section. The section is pulled by a drive in some portions of its path and pushed in other portions. In inclined portions of the track, where a higher driving force is needed, the drives are arranged more closely than in horizontal portions. The drives are put in operation periodically one after another as a train approaches them. Conveyer trains are known with drives arranged on the train sections, but they have only a limited application.

The motion of train sections may be reciprocating ('shuttle' motion) when loaded sections move in one direction and unloaded ones, in the opposite direction on the same rail track (open path, Fig. 17.1*a*), a continuous along a circular path which may be horizontally closed (Fig. 17.1*b*), vertically closed (Fig. 17.1*c*) or spatial (Fig. 17.1*d*). A closed circular system has separate tracks for loaded and empty train sections; the tracks can be arranged in one stage, two stages (one above the other) or combined. Various branches and sideways with transfer switches *Sw* may be provided. A train section may be loaded and unloaded in motion by means of loading (*LD*) and unloading devices (*UD*) whose design is determined by the type of train section and the characteristics of the load.

The motion of train sections can be controlled locally or centrally, with automatic distribution of sections to various tracks.

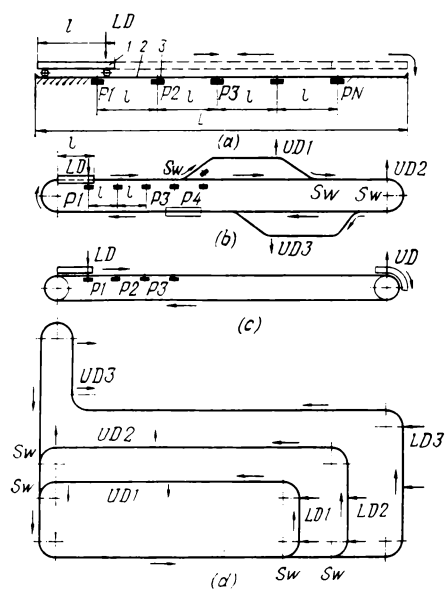


Fig. 17.1. Schemes of paths of conveyer trains

(*a*) open; (*b*) circular horizontally closed; (*c*) circular vertically closed; (*d*) composite spatial

A system of conveyer trains can be characterized by cyclic transportation of loads. In that respect it differs radically from conveyers which are characterized by continuous motion of loads. The cyclic operation of conveyer trains follows from their principal design and is also dictated by the necessity to slow down the sections at loading and unloading stations. For that reason, the throughput capacity of conveyer trains is under comparable conditions always lower than that of a continuous conveyer. Conveyer trains operate on the same principle, but have numerous design versions.

The principal characteristics of conveyer trains in industrial plants are as follows: throughput capacity 500-1500 t/h; transportation distance 4-23 km; highest slope 22°; largest lifting height 500 m; speed at loading and unloading stations not more than 2 m/s and in transportation up to 7 m/s; length of trains up to 330 m; the number of trains in a system 8-22; mass of transported load per metre of train section length 140 kg; turning radius in horizontal plane 100 m and in vertical plane 20 m; transported load: rock lumps of a size 0-350 mm; application: in ore quarries.

The advantages of conveyer trains are the possibility of track branching to deliver various loads to various unloading stations in a system; wide range of materials suitable for transportation, including large-lump loads (provided that the load-carrying part of train is designed properly); high angles of incline (up to 30°); the possibility of load conveying on complex bent paths with vertical and horizontal turns; and a certain accumulating capacity.

Among the drawbacks of conveyer trains are complicated design and large length of unloading stations for side unloading; the necessity to arrange a large number of drive units on the train path whose total installed power is substantially higher and the reliability, much lower than with continuous-action conveyers; high resistance and increased power consumption at starting; reduced throughput capacity owing to cyclic mode of load transportation; and complex design and poor reliability of the existing types of drives for conveyer trains.

Conveyer train systems are being developed for the application in open-cast quarries. A number of experimental conveyer trains have been designed in this country, but their industrial implementation will become possible when an optimal technological scheme of their application is found.

17.2. Elements of Conveyer Trains

The moving part of a train section, as a rule, consists of a number of single-axle carriages hingedly connected with one another (Fig. 17.2*a*) and a load-carrying element placed on the carriage frames. Owing to hinged connection of carriages, a train section has spatial flexibility, i.e. can move on spatial paths with horizontal and vertical turns.

The load-carrying element is in the form of a corrugated belt as in spatial apron conveyers (see Fig. 17.4), troughed steel floor (Fig. 17.2*a*), railway carriage boxes with or without end walls or other containers for bulk loads, which are connected with one another by elastic elements (say, sections of rubberized belt) to prevent fall-through of the material between carriages in continuous loading of a moving train.

The tractive element of a train section may be in the form of longitudinal beams of the carriages, sections of round-link chain, steel cables, etc.

The drive is of the caterpillar type with engagement by the moving parts of a train section (Fig. 17.3); a friction drive with driving pneumatic-tyre wheels arranged horizontally (see Fig. 17.2) or vertically; a friction drive with

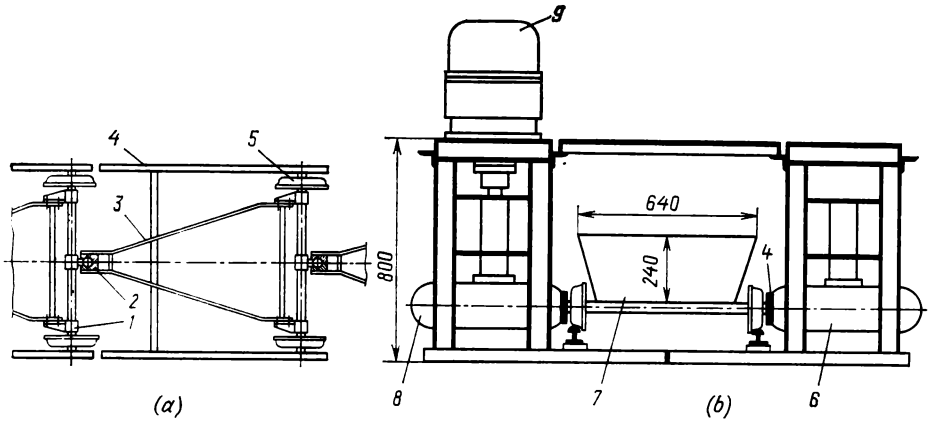


Fig. 17.2. Conveyer train type SECCAM (France) with single-stage arrangement of conveyer paths

(*a*) moving part of conveyer (*b*) friction drive with horizontal driving wheels; *1*—car axle; *2*—spherical joint; *3*—car frame; *4*—longitudinal beams; *5*—wheel; *6* and *8*—driving wheels; *7*—load-carrying box; *9*—drive mechanism with electric motor

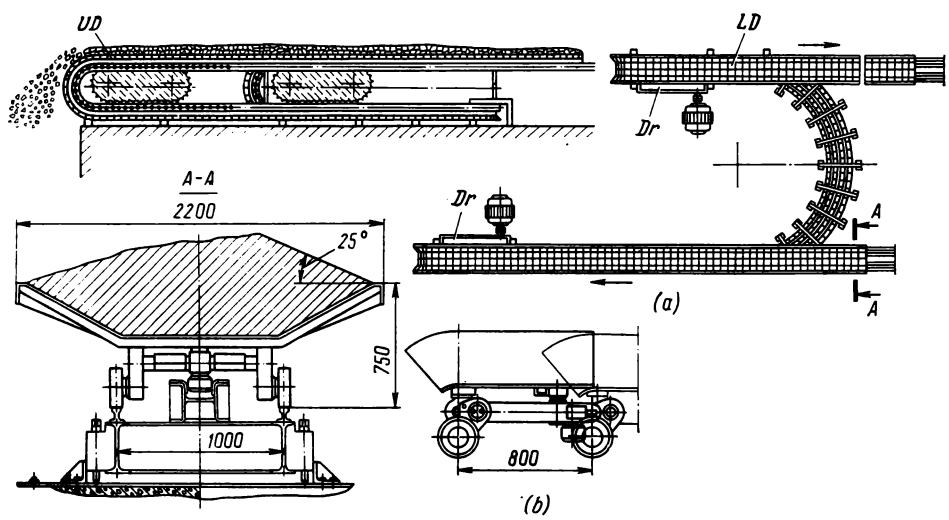


Fig. 17.3. Conveyer train with combined arrangement of paths

(*a*) unloading (*UD*) and loading (*LD*) devices; *Dr*—caterpillar drive; (*b*) moving part of conveyer

longitudinal rubberized belts and pressure rollers; magneto-frictional caterpillar drive; induction drive with linear asynchronous motors (Fig. 17.4); etc. The problem of finding the optimal drive for conveyer trains still has to be solved. Drives with linear asynchronous motors seem to be most promising.

The loading device usually consists of a bin and a feeder which delivers the material from the bin continuously (or intermittently) onto a moving train section. The speed of train sections in loading stations is usually 0.5-2 m/s.

The unloading device is usually of the end type (over the conveyer end), as in common apron conveyers (see Fig. 17.3). In sideway unloading, the boxes of containers of a train section are tilted or turned over on the longitudinal axis. With sideway unloading, the car-

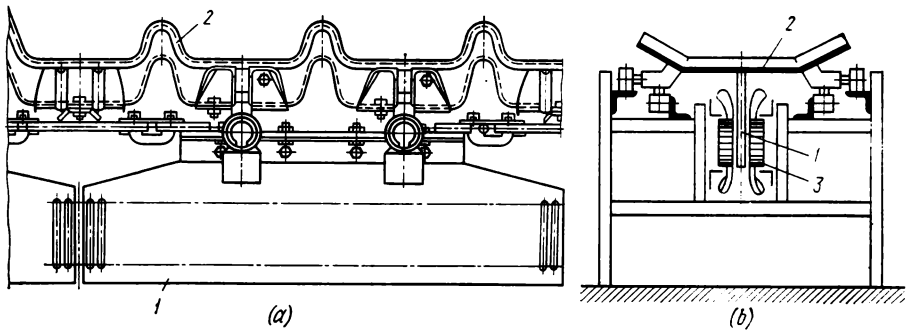


Fig. 17.4. Conveyer train with single-stage arrangement of tracks
(*a*) moving part; (*b*) cross section through linear asynchronous motor drive; *1*—secondary part of motor ('rotor'); *2*—load-carrying corrugated belt; *3*—primary part of motor ('stator')

rying and guide tracks should be bent along a spiral so as to lift and turn over the carriage boxes and then return them into the initial position. The length of the tracks of an unloading station of this type is 30-60 m. Additional forces appearing in track bends cause intensive wear of the tracks and moving parts of train sections.

17.3. Calculation of Conveyer Trains

The throughput capacity (t/h) of a conveyer train

$$Q_1 = 0.36 q_l l_s K_n / (T_1 K_t K_a) \qquad (17.1)$$

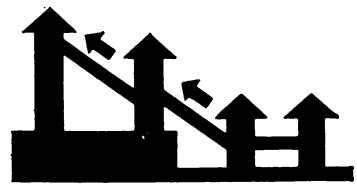
where q_l is the linear gravity force of the load transported on the load-carrying element of train section, N/m; l_s is the length of train section, m; K_n is the coefficient of non-uniformity of operation ($K_n = 1.2$); $K_t = 0.8\text{-}0.9$ is the coefficient of utilization in time; $K_a = 0.90\text{-}0.95$ is the coefficient of availability of the system; and T_1 is the time of a cycle of train motion, s.

The time of a cycle of train motion

$$T_1 = \frac{L_l - (l_l + l_u)}{v_t k_{ac}} + \frac{L_0}{v_0} + \frac{l_l}{v_l} + \frac{l_u}{v_u} \qquad (17.2)$$

where L_l and L_0 are the lengths of the load-carrying and return track, m; l_l and l_u are the lengths of the loading and unloading stations, m; v_t and v_0 are the speeds of transportation in the load-carrying and return runs, m/s; v_l and v_u are the speeds in the loading and unloading stations, m/s; and $k_{ac} = 0.8\text{-}0.9$ is the coefficient which considers acceleration and braking of the train before and after loading and unloading stations.

18
Typical Applications of Conveying Machines in Complex Systems



18.1. General Concept of a Completely Mechanized and Automated System

In modern industrial plants, materials transport and handling is mechanized by using a system consisting of a number of conveying machines and devices which operate in a coordinated mode with the processing equipment. Such systems become most efficient and perfect in engineering embodiment when they contain a whole complex of machines and appliances for mechanized transportation of loads along the entire length of cargo traffic from the first to the last process operation both in the main and auxiliary sections of the production process. Manual labour then can be reduced to controlling machines and performing certain minor operations and planned repairs of the equipment. This higher form of mechanization is called 'comprehensive mechanization'. If only a single auxiliary operation is not mechanized completely, the efficiency of operation of the whole machine complex may reduce substantially.

The highest stage of comprehensive mechanization is comprehensive automation of individual technological sections and of the whole production process. In comprehensive automation, the operation of the entire complex of transport and process machines in a production process is controlled automatically by means of various instruments and control devices; in some complicated production processes, automatic control is effected by electronic computers. Man's interference is then only needed to set a control programme for a machine complex, adjust the machines and controls, check their operation, and carry out planned repairs of the equipment and controls.

Comprehensive mechanization and automation are employed widely in modern plants in various branches of industry in this and other countries. Examples are known of successful operation of fully automatic bakeries, concrete-mixing plants, automatic flow lines for the manufacture of pistons and piston rings, etc.

At the present level of mechanized transportation, all modern plants have numerous conveying machines. Below are discussed some typical examples of application of conveying machines and auxiliary devices in completely mechanized process-transportation systems. More detailed data on such systems may be found in special teaching courses.

18.2. Mechanized Transport and Handling in a Foundry Shop

Foundry is characterized by intensive cargo traffic. Metal, fluxes and fuel should be transported and handled in

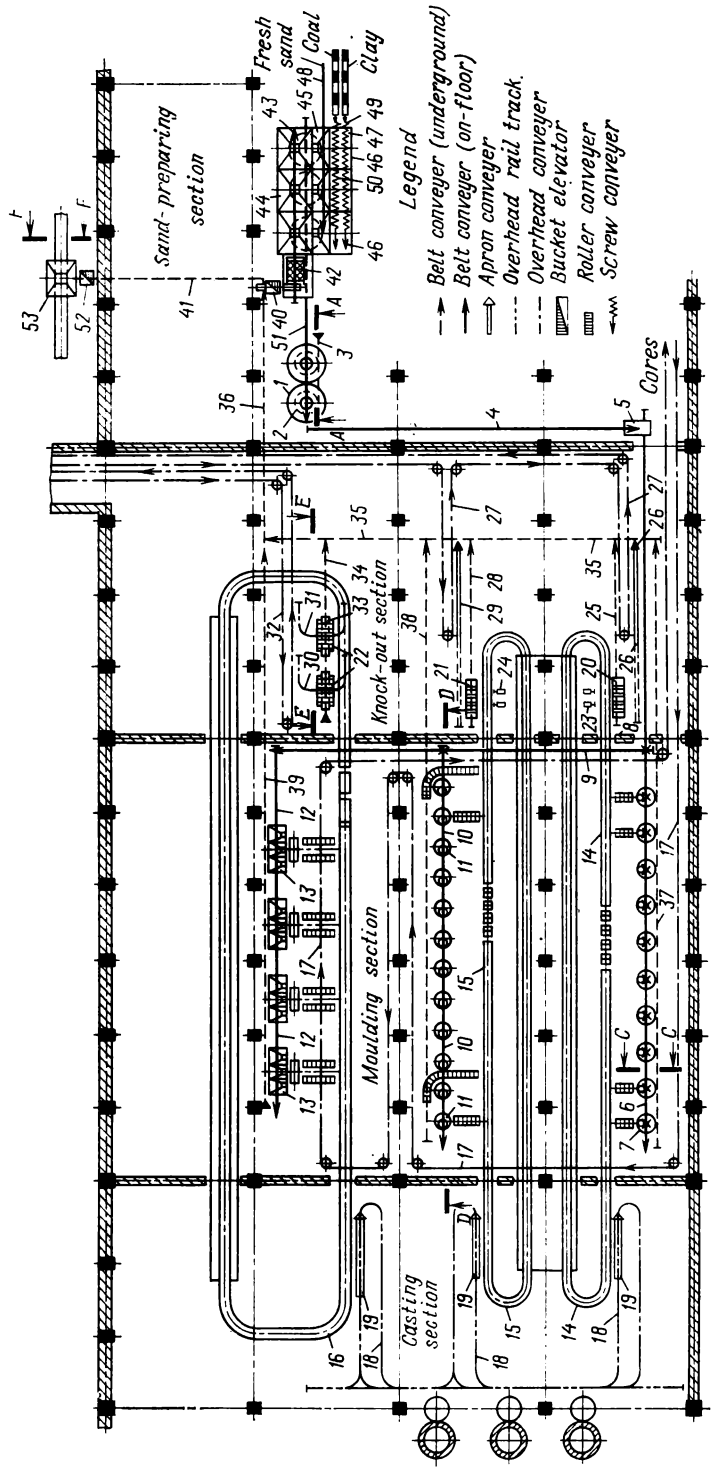


Fig. 18.1. Layout of conveying machines in a foundry shop

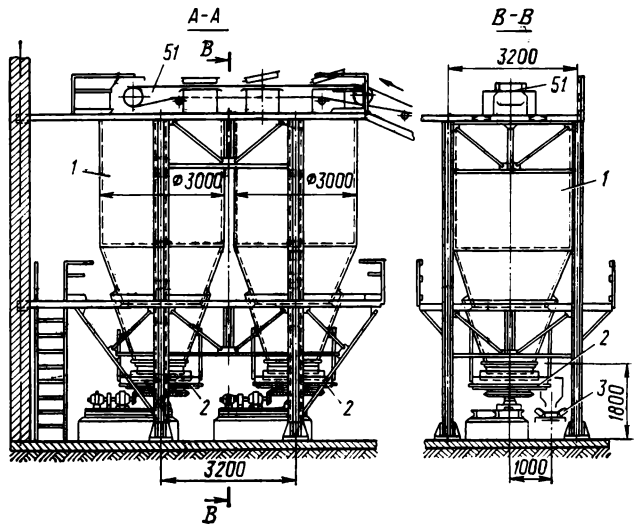


Fig. 18.2. Settling bins for prepared foundry sand (section *A-A* in Fig. 18.1)

the stockyard and melting section; foundry sand, cores, and moulds in the moulding section; sand, binders and cores in the core-making section; used and fresh sand, coal and clay in the sand-preparing section; sand, coal and clay in the drying section; castings and wastes in the chipping section. Depending on the scale of production, from 65 to 200 t of various loads should be transported in a foundry shop per ton of sound castings, i.e. the total intensity of load transportation in the shop is 120-600 t/h on the average. The layout of conveying machines in a mechanized foundry shop which makes small- and medium-size grey-iron castings for a machine-building works is shown in Fig. 18.1. Let us consider the main paths of cargo traffic in the shop.

Prepared moulding sand in settling bins *1* (Fig. 18.2) is taken by a disc feeder *2*, with the disc having a diameter of 1.5 m, and fed onto a belt conveyer *3* (belt width, 650 mm; and belt speed, 1 m/s). The sand is then reloaded onto a conveyer *4* (see Fig. 18.1), passes through an aerator *5*, is carried in the loosened state ly a conveyer *6* (see Fig. 18.3), and distributed between the bins of a bin trestle above moulding machines. The sand is discharged from the conveyer *6* into bins of the trestle *7* by means of automatically controlled plough-type unloaders. For delivery to other trestles, sand from the conveyer *6* is scraped off by a plough-type unloader *8* (see Figs. 18.1 and 18.4) onto a belt conveyer *9* (Fig. 18.5). Sand is discharged from this conveyer by a plough-type unloader onto a conveyer *10*

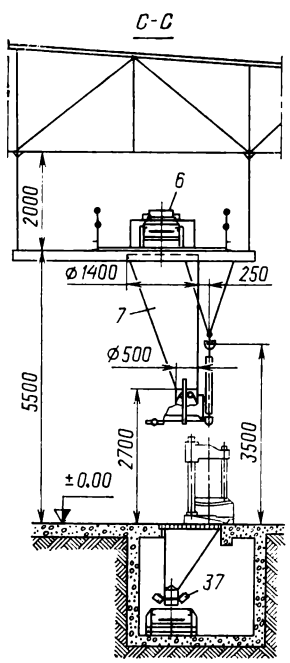


Fig. 18.3. Cross section through bin trestle above moulding machines (section *C-C* in Fig. 18.1)

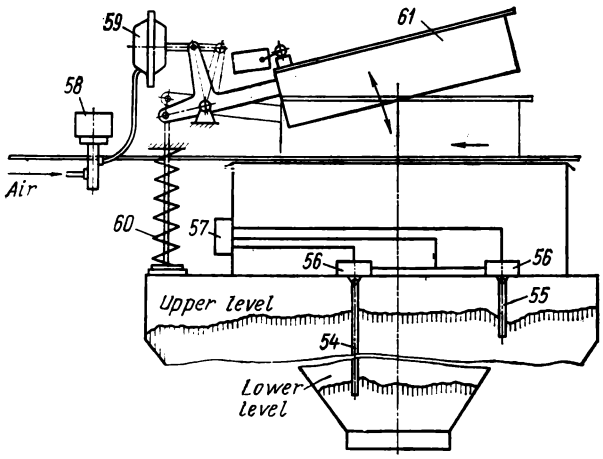


Fig. 18.4. Automatic plough unloader of foundry sand into bins

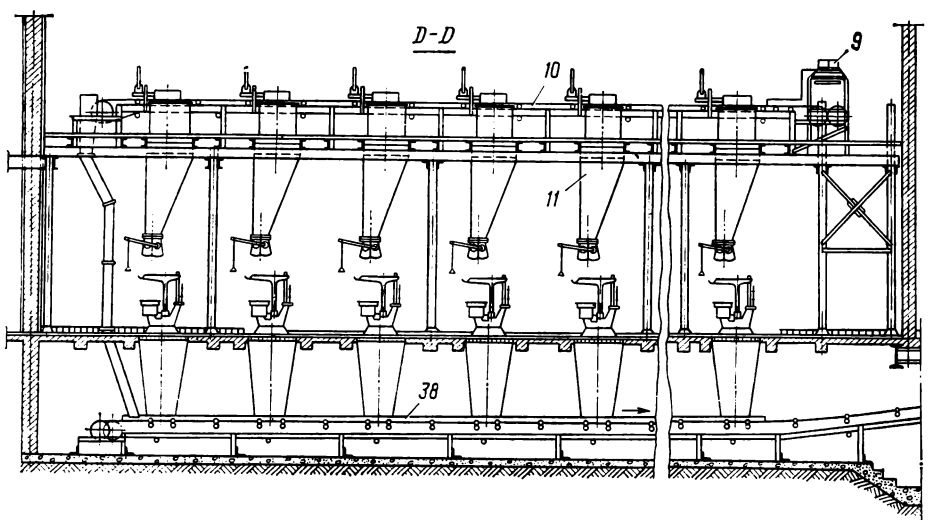


Fig. 18.5. Distributing belt conveyer and bin trestle above moulding machines (section *D-D* in Fig. 18.1)

of a bin trestle *11* above the moulding machines or directly over the end pulley onto a conveyer *12* (see Fig. 18.1) of the bin trestle *13* above the heavy-type moulding machines.

Sand between the bins is distributed by means of plough unloaders (see Fig. 18.4) which are put into operation automatically depending on the sand level in the bins. A plough unloader is switched on by a control signal supplied by an automatic level gauge which measures the lowest level in the bin. The plough unloader is switched off by a signal from another level gauge which measures the highest level in the bin.

A level gauge has rod electrodes *54* and *55* (see Fig. 18.4) connected with sensors *56* and a control relay *57* with an electromagnetic air valve *58* and a pneumatic membrane chamber *59* which together with a spring-biased mechanism *60* can lift and lower the plough *61* of an unloader.

There are a number of automatic systems for successive operation of plough unloaders on a conveyer which supplies bulk material to a group of bins. All bins are provided with jaw-type

gates actuated by pneumatic cylinders. Prepared mould halves are carried by pneumatic lifters (which move on overhead tracks) onto roller conveyers where they are assembled into moulds and transferred onto the carriage conveyers *14*, *15*, and *16* (see Fig. 18.1).

Cores are delivered from the core-making section to the lines of moulding machines by means of an overhead conveyer *17*. Prepared moulds are carried by car conveyers into the casting bay. Molten iron from cupola furnaces is poured into casting ladles with tilting mechanisms. The ladles are suspended from electrically driven carriages which move on overhead two-rail tracks *18* to conveyers for carrying casting moulds. In the casting section, a mobile casting platform *19* is provided at each conveyer, the platform being essentially an apron conveyer with a flat metallic floor 650 mm wide and 30 m long, which moves at the same speed as the mould-carrying conveyer. The workers stand on the moving platform and pour the metal from ladles into the moving moulds on the conveyer cars.

Filled moulds are transported by the car conveyer through cooling sections (where the moulds are cooled) to knock-out grids *20*, *21*, and *22*. Moulds with castings are pushed off from the conveyers *14* and *15* by pneumatic pushers *23* and *24* onto knock-out grids *20* and *21*, where the moulds are disassembled. In the section of conveyer *14*, knocked-out sand falls through the grid onto an underfloor belt conveyer *25* with a heat-resistant rubberized belt. Hot castings move along a slide into the basement onto an apron conveyer *26* moving at a speed of 0.16 m/s; the apron is made from steel plates and has side walls, its width is 800 mm.

Empty mould halves are placed onto platforms of the casting conveyer (or onto a roller conveyer with a pusher) and returned to the moulding machines. Foundry plants are known in which heavy and labour-consuming knock-out operations are performed automatically.

Hot castings are lifted from the basement by the apron conveyer *26*, reloaded semi-automatically (or automatically) onto an overhead cooling conveyer *27* and transported on this conveyer into the chipping section; the castings are cooled during the transportation. Similar operations are performed in the line of conveyer *15* by means of a belt conveyer *28* and apron conveyer *29*. Hot castings are reloaded from conveyer *29* onto the overhead cooling conveyer to be transported into the chipping section.

In the knock-out section of the casting conveyer *16*, moulds are reloaded from the conveyer onto knock-out grid *22* by means of pneumatic lifters (or electric hoists) which move on overhead tracks *30* and *31*. Empty moulds are returned on a roller conveyer to the moulding machine line, hot castings are hung onto suspensions of an overhead cooling conveyer *32* and carried to the chipping sections.

Knocked-out sand falls onto apron feeders *33* (Fig. 18.6) with a steel-plate floor of a width of 650 mm and with side walls (apron speed 0.16 m/s), which are mounted under the knock-out grids *22*. The feeders *33* supply the sand onto a belt conveyer *34* in the basement. Used sand is transferred from longitudinal conveyers *25*, *28* and *34* (see Fig. 18.1) onto a transverse belt conveyer *35* and from this onto a belt conveyer *36*. Underground belt conveyers *37*, *38* and *39* are arranged along each moulding machine line to collect the sand which falls from distributing conveyers *6*, *10* and *12* during filling of the bins. The knocked-out and collected sand is carried by a belt conveyer *36* into the sand-preparing section to a bucket-belt elevator *40* with deep widely spaced buckets of a width of 320 mm fastened to the belt 400 mm wide; the belt moves at a speed of 1.25 m/s. The reloading station is designed so that the sand discharged over the drive pulley of conveyer *36* can be directed by a turning valve in the

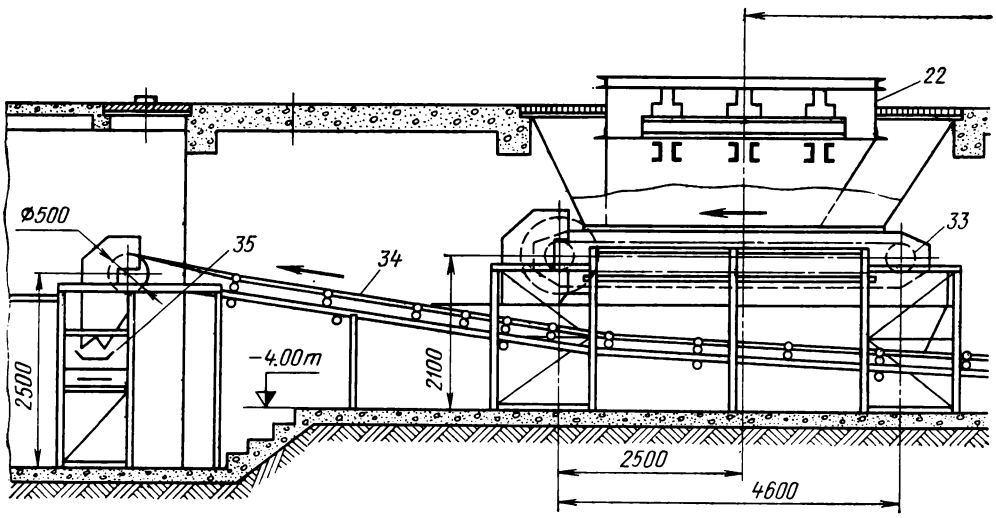


Fig. 18.6. Delivery of sand from beneath knock-out grates (section *E-E* in Fig. 18.1)

funnel either into the charging nose of elevator *40* to be returned into the sand-preparing section, or onto a belt conveyer *41* to be rejected to a dump.

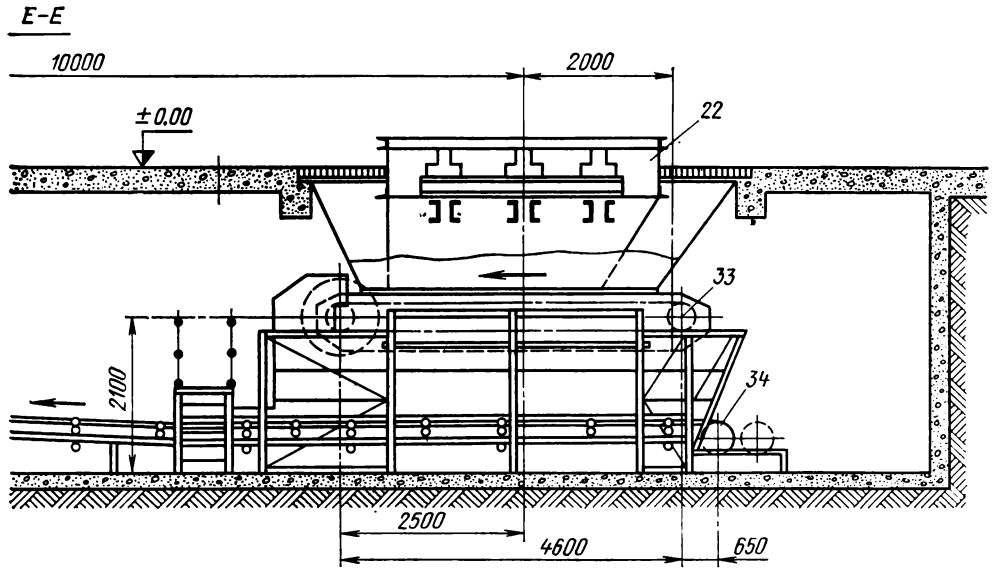
The drive pulley of conveyer *36* has magnets inside, so that iron particles are separated from sand.

The bucket elevator *40* raises the sand to a bin trestle above mixing runners and discharges it into a cylindrical sieve *42*. The screened sand is fed onto a distributing belt conveyer *43* from which it is discharged by plough-type unloaders into used-sand bins *44*. There are three bins above each pair of runners: a bin *44* for knocked-out (old) sand (of a volume of 15 m^3), a bin *46* for pulverized clay (3 m^3), and a bin *47* for pulverized coal (2 m^3).

Fresh sand is delivered from a sand drier by a belt conveyer *48* and distributed between bins by plough-type unloaders. Pulverized clay and coal are transported from the sand drier by pneumatic transport and distributed between bins by tightly sealed screw conveyers *49* and *50* with screws 200 mm in diameter. Old sand, fresh sand, clay and coal are fed by meters in required quantities into the mixing runners where they are wetted and intermixed automatically. The prepared sand mixture is discharged from the runners onto a belt conveyer *51* (see Fig. 18.2) and carried to settling bins *1* each of a capacity of 30 m^3, from which it can be delivered to the moulding section.

The surplus of knocked-out sand from conveyer *36* (see Fig. 18.1) is fed, on readjustment of the turning valve in the funnel, onto the underground belt conveyer *41* (Fig. 18.7), which carries it to a bucket elevator *52* with widely spaced deep buckets. The used sand from the elevator *52* is discharged into a bin *53* (of a volume of 70 m^3) mounted above a railway track. The bin is unloaded periodically into a train of railway cars. The entire system of conveyers is operated and controlled from the central control panel.

We have described only the main material flows in sections of a foundry shop. Additionally, a large number of conveying machines operate in the core-making section (overhead, cradle and



belt conveyers), sand-drying section (belt conveyers, belt-bucket elevators, disc feeders, pneumatic transport means), stockyard (belt conveyers for delivery of limestone and coke, roller conveyers for delivery of charge boxes, etc.), and in the chipping section (overhead, apron, roller conveyers, etc.).

The operation of a modern foundry shop for series or mass production would be practically impossible without conveying machines.

18.3. Mechanized Transport and Storage in the Core-making Section of Foundry Shop

The core-making section of the foundry shop of an automobile works is arranged in a three-floor building. Cores are moulded, dried and stored on the third floor, assembled into sets on the second floor, and placed into moulds on the first floor where the moulds are filled with metal. All operations of transport and storage are performed by a system of three overhead pusher conveyers *C1*, *C2*, *C3* (Figs. 18.8, 18.9) of the same design, with cars of a load-carrying capacity of 320 kg. The conveying system transports 30 000 various cores during a shift.

The load-carrying attachments are made in the form of three-stage stacks with shelves 1200 mm × 900 mm; the pitch of attachments in the main conveyer lines *C1* and *C2* is 2.4 m, in auxiliary storage conveyers 1.2 m, and in the conveyer *C3*, 1.8 m. Conveyer cars are provided with key-type address carriers of an electromechanical automatic addressing system for 16 addresses. The tractive element of the conveyers is a detachable chain with a pitch of 100 mm; the spacing between cars is 600 mm and the length of the conveyers respectively 400 m and 500 m. The chain speed is 8.4 m/min in the conveyers *C1* and *C2* and 12 m/min in the conveyer *C3*.

The conveyer *C1* (Fig. 18.8) carries cores from the line of moulding machines *MM1* to a drying kiln *DK1* (which is provided with an auxiliary conveyer *AC1*) and to a store *St1*, where the cores are cooled and distributed according to their types between sideways. The store contains a certain reserve of cores required for the independent operation of core-assembling lines and for

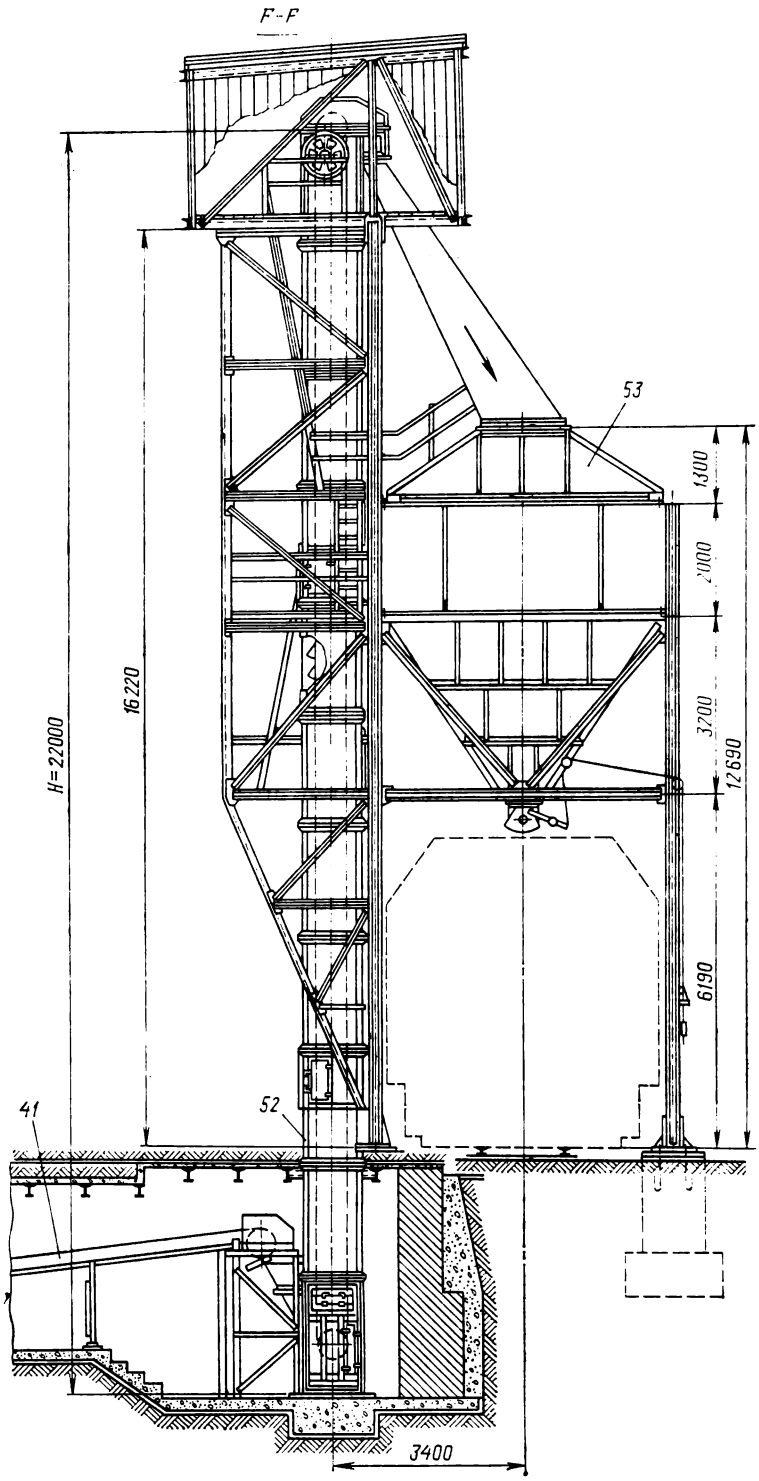


Fig. 18.7. Elevator for lifting used sand into a bin above railway track (section *F-F* in Fig. 18.1)

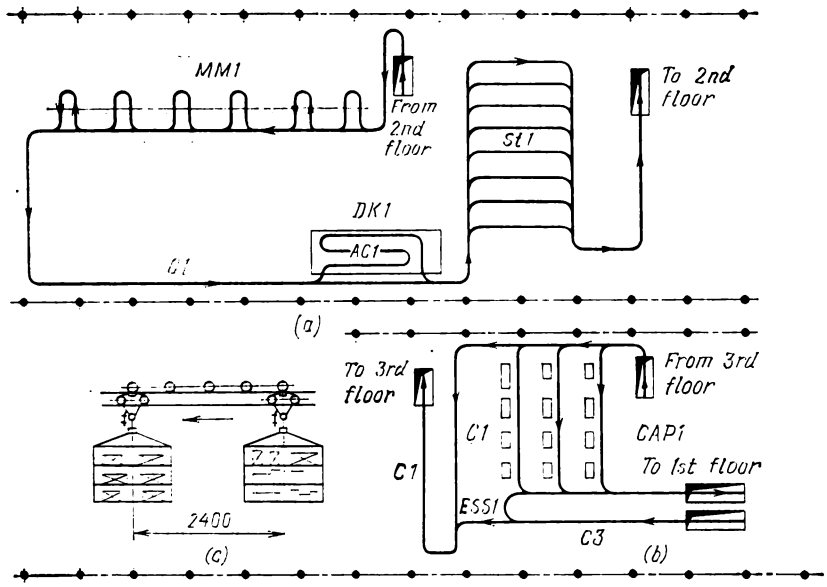


Fig. 18.8. Schematic of overhead pusher conveyer *C1* for mould cores
(a) third floor; (b) second floor; (c) conveyer section with load-carrying attachments

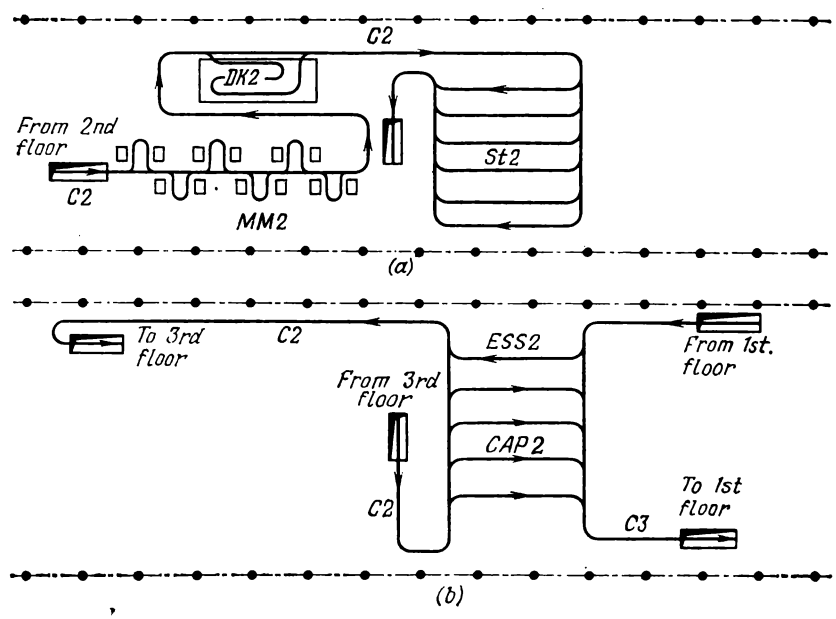


Fig. 18.9. Schematic of overhead pusher conveyer *C2* for mould cores
(a) third floor; (b) second floor

the delivery of cores to be placed into casting moulds. The capacity of the store is up to 600 load-carrying attachments. An address for a sideway switch of the store is set up by a worker in the moulding machine line. Load-carrying attachments are moved on tracks by means of auxiliary conveyers.

Cars with the required cores are moved, by operator's command, from the store tracks through outlet switches again into the path of conveyer *C1* which carries them onto the second floor to sideways at core-assembling places *CAP1*. Upon passing the sideways of *CAP1*, the empty run of the conveyer returns to the third floor. The attachments with assembled cores are addressed by workers at *CAP1*, pass through an outlet switch into the conveyer line *C3*, and brought onto the first floor, the cores are placed into moulds. Cars with empty attachments are moved from the conveyer *C3* through an empty-attachment section *ESS1* to the conveyer *C1*.

The conveyer *C2* (Fig. 18.9) performs the same transport operations as the conveyer *C1*, but with cores of other types. It carries the cores from the moulding-machine line *MM2* to a drying kiln *DK2* and a store *St2* and hence onto the second floor to core-assembling places *CAP2* and returns to the third floor. The attachments with assembled cores are carried from *CAP2* to the conveyer *C3* and transported to the first floor to working places where cores are placed into moulds. Empty attachments pass through a section *ESS2* onto the conveyer *C2* and return into the line *MM2*.

18.4. Mechanized Transport in a Fuel-supply Shop of a Thermal Power Station

The intensity of fuel flows at thermal power stations may be as high as 500 t/h or even more. The problem of fuel transport and handling can be practically solved only by complete mechanization beginning from the delivery of fuel to a station and up to its feeding into the bins and furnaces of boilers. Continuous operation of a station requires a 24-hour reliable and trouble-free operation of conveying machines and process plants. To ensure this, the main conveying lines in fuel-supply shops of thermal power stations are usually doubled and operate in succession: when one line is operating, the other stands as reserve.

Schematics of conveyer layout for fuel supply to a regional thermal power station consuming around 400 t/h of coal are shown in Figs. 18.10 and 18.11.

Coal is delivered to the station by trains of dump-cars; a train may be moved directly onto the track *1* of the receiving device of the fuel-supply system or the track *2* of a reserve fuel store from which the fuel can be taken at intervals between trains. A train on the track *1* is unloaded into slotted bins *3* arranged under the rail track at both sides along the entire unloading front. Blade-type unloaders *4* and *4a* arranged under the slotted bins discharge the coal onto a belt conveyer *5* or *5a* (when one of them operates, the other stands by). Reloading trays are arranged so that coal can be reloaded from the conveyer of one line onto the next conveyer of the same or adjacent line. Coal is reloaded from the conveyer *5* onto a horizontal-inclined belt conveyer *6* which carries it to a coal-crushing section. Before the coal is discharged from the conveyer *5* into coal crushers, suspended or drum-type magnetic separators remove iron particles from it.

Crushed coal from the crushers is discharged onto a horizontal-inclined belt conveyer *7* with automatic scales *8* for continuous weighing of coal. At a next reloading station, there is a chip separator *9* which removes wooden inclusions from the coal. Coal is discharged from the conveyer *7* onto a distributing belt conveyer *10* mounted above bins *11* of the boiler house.

An automatic drum-type unloader *12* movable along the conveyer *10* above the bins discharges the coal from this conveyer into the boiler house bins, from which it can be fed by special devices into boiler furnaces.

When the bins of the receiving arrangement are filled with coal, a tra-

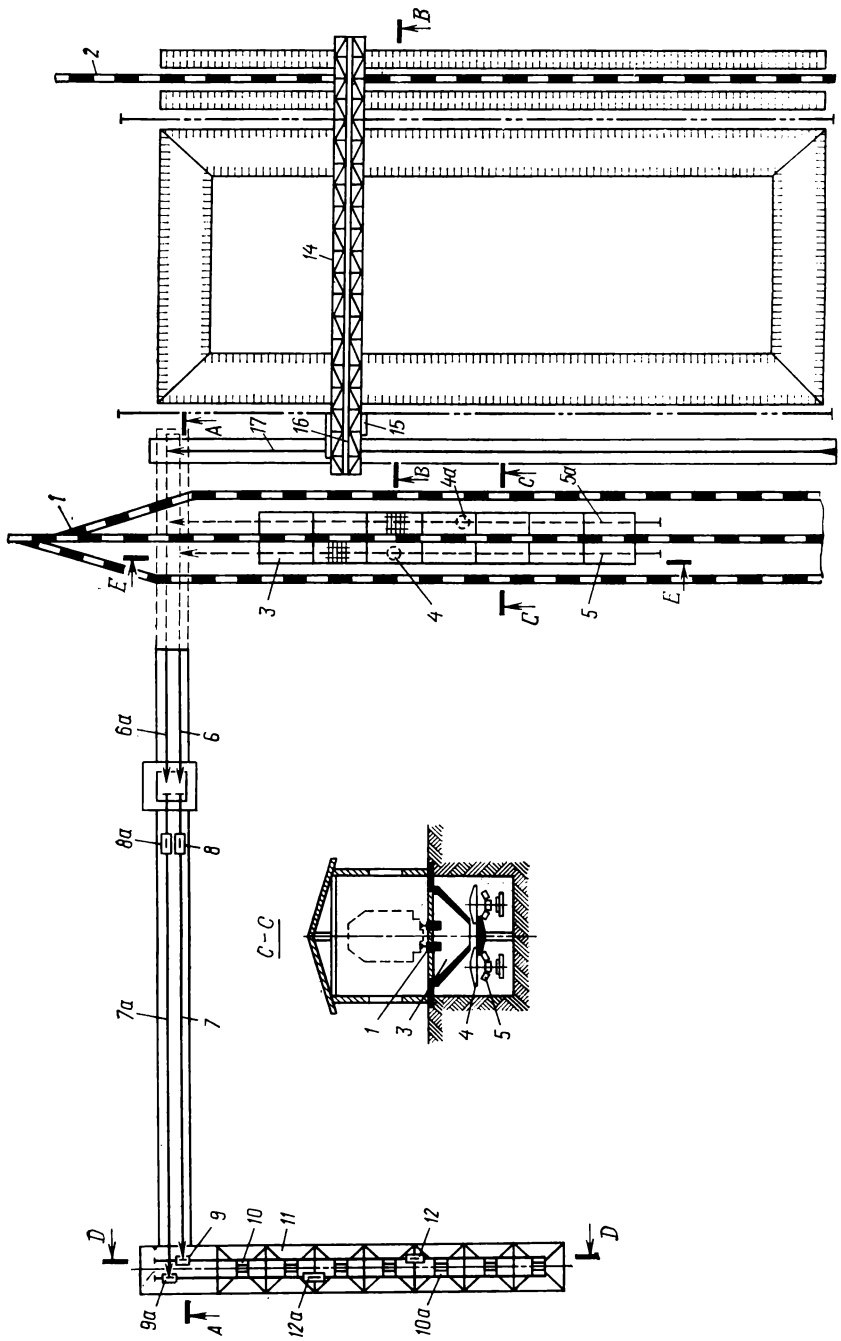


Fig. 18.10. Layout of conveying machines of fuel-supply system at thermal power station

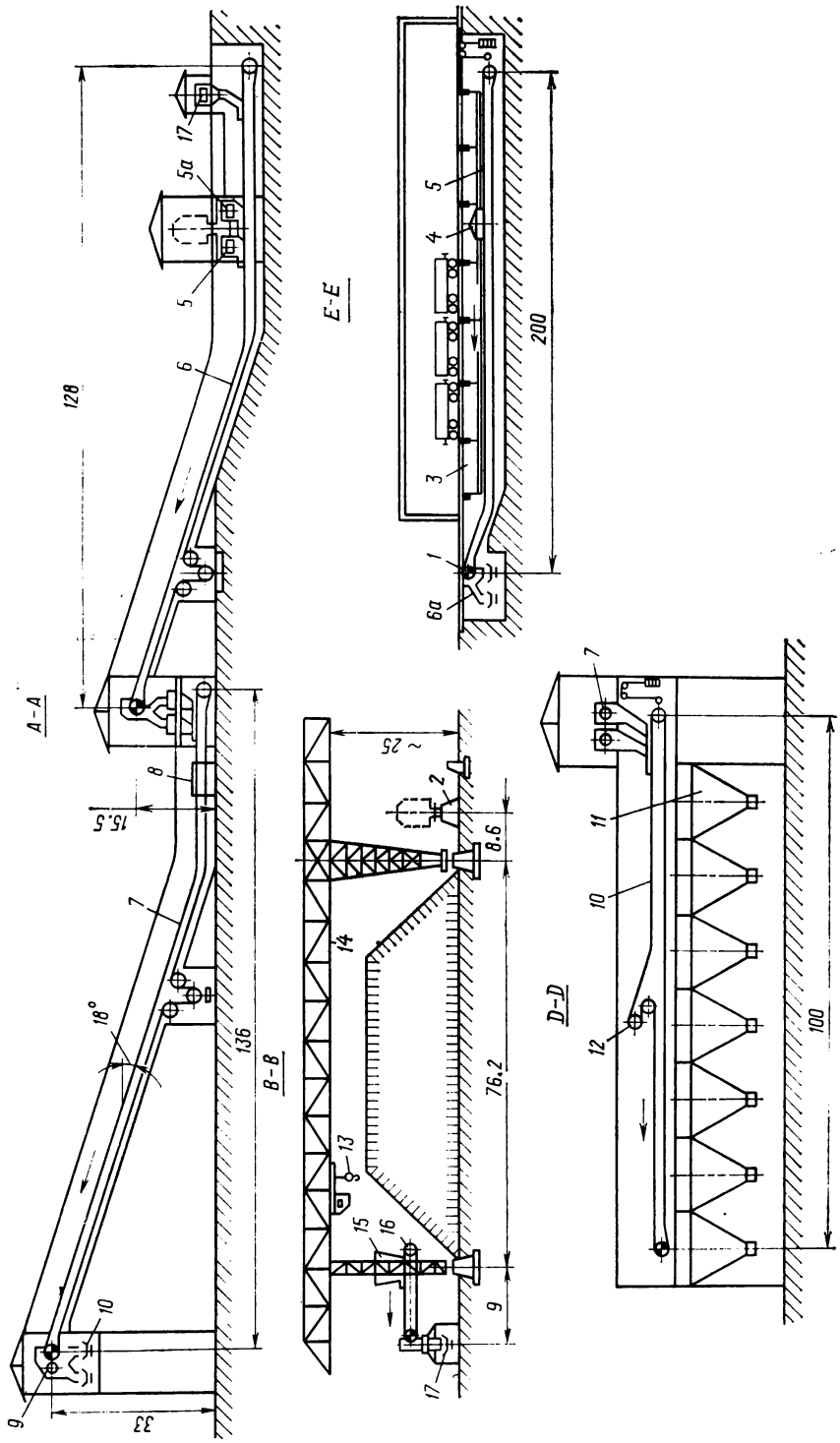


Fig. 18.11. Cross sections of conveying machines of fuel-supply system (to Fig. 18.10, dimensions in metres)

in with coal is moved onto track *2* on an open trestle with two trenches along the track. Coal is discharged from dump-cars into the trenches and then can be taken by the grab *13* (see Fig. 18.11) of a gantry reloader *14* and piled in an open store which usually contains a coal reserve for three months of operation.

To supply fuel from the store to boilers, coal is taken from a pile by the same grab *13* and carried into a bin *15* mounted on the frame of the reloader. An apron feeder *16* mounted on a cantilever arm under bin *15* discharges the coal through openings in the trestle cover onto a belt conveyer *17* running along the entire length of the coal store. The conveyer *17* delivers the coal to the conveyer *6* of the main fuel-supply line. All belt conveyers of the fuel-supply system have a rubberized belt of a width of 1000 mm moving at a speed of 1.6 m/s on roller supports.

The conveyer system is controlled from a central panel which ensures automatic successive starting and stopping of the conveyers. The panel has a mnemonic diagram of the conveyer system with all controlling sensors. If a trouble occurs, the system is stopped by the signal of a particular sensor and this is indicated on the control panel.

18.5. Mechanized Transport at a Stone-crushing Mill

Civil engineering and road construction require large quantities of crushed stone which can be produced in powerful mechanized stone-crushing mills.

A mill has a complex of conveyers of a total throughput capacity 1000 t/h (Fig. 18.12). Stone is delivered from an excavator face to the mill by a belt conveyer *1* of a length of 500 m with a belt 1200 mm wide running at a speed of 2 m/s. The conveyer has automatic scales *2* which automatically measure the quantity of stone supplied to the mill. Stone is discharged from the conveyer *1* into a primary crusher *3*. Crushed stone of a size less than 180 mm is discharged from the primary crusher *3* into a bin *4* of a volume of 50 m^3 whence it is delivered by a shaking feeder *5* onto an inclined belt conveyer *6* of a length of 80 m (belt width 1400 mm, speed 1.6 m/s) and then reloaded onto a reversible mobile ('shuttle-type') conveyer *7* which distributes the load over a pile *8* of a volume up to 50 000 m^3. The pile serves as a buffer store to ensure continuous operation of the mill at interruptions in stone delivery from the face or if the mill

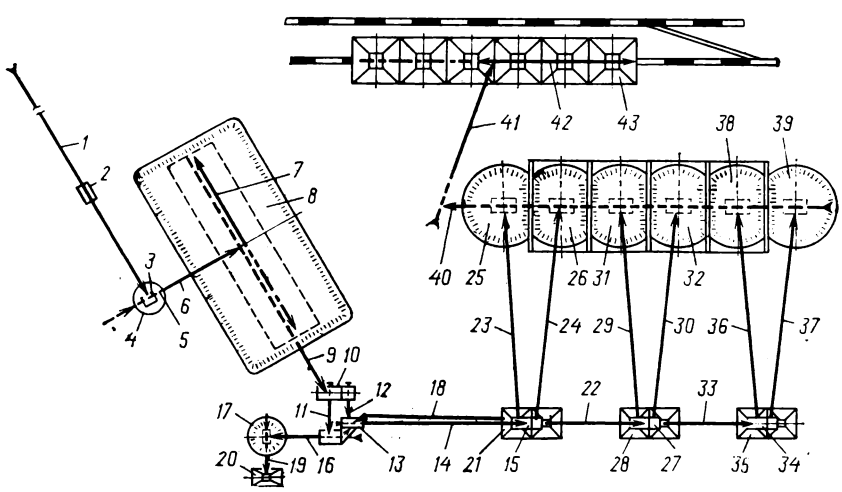


Fig. 18.12. Layout of conveying machines of stone-crushing mill

operates on a two-shift schedule and the excavator face, on a single-shift.

A tunnel with three bins is arranged under the pile. Shaking feeders of a throughput capacity of 1000 t/h, which are mounted under the bins, operate successively and reload the material onto a belt conveyer *9* of a length of 105 m which carries the stone to a crushing and sorting mill *10* where stone is crushed additionally and graded into sizes. The fraction from 0 to 100 mm is fed onto a belt conveyer *11* (belt width 1000 mm, and length 45 m) and the fraction from 0 to 80 mm, onto a conveyer *12* (belt width 1000 mm) which returns the stone for repeated grading and crushing into a mill *13* after which the material is carried by a belt conveyer *14* (belt width 1000 mm) into an additional screening plant *15*.

The material supplied by the conveyer *11* is also crushed and graded, with large-size gravel (25-50 mm) being carried by a conveyer *16* (belt width 650 mm, and speed 1.6 m/s) into a pile *17* of a volume of 900 m^3, while fine gravel (up to 25 mm) is reloaded onto the conveyer *14*. A tunnel is provided under the pile *17*; it has a charging bin from which stone is discharged by a shaking feeder onto a belt conveyer *19* (length 22 m, width 650 mm, and speed 1.6 m/s) and carried into a charging bin *20* to be loaded into tip-lorries and delivered to consumers.

Gravel is graded in the grading plant *15* into three grades: 0-20 mm; 20-30 mm, and 30-50 mm. The two larger fractions are fed into charging bins *21* each of a volume of 200 m^3, and the fraction 0-20 mm is discharged onto a belt conveyer *22*. Gravel from the bins *21* is reloaded into tip-lorries to be delivered to consumers. When the bins *21* are filled, stone from the plant *15* is fed directly onto belt conveyers *23* and *24* (length 50 m, width 650 mm, speed 1.6 m/s) to be stored in piles *25* and *26*.

Stone lumps of a size more than 50 mm are returned from the plant *15* by conveyer *18* for repeated crushing. Conveyer *22* carries the material into a crushing and grading plant *27* which makes three grades of gravel: 0-12 mm, 12-16 mm, and 16-20 mm. The two larger fractions are fed into bins *28* from which they can be unloaded into tip-lorries. When the bins *28* are filled, gravel of these two grades is carried by conveyers *29* and *30* into piles *31* and *32*. The fraction 0-12 mm is carried by a belt conveyer *33* to a screening plant *34* for final grading into two fractions: 0-6 mm and 6-12 mm. These grades of gravel are charged into two bins *35* each of a volume of 20 m^3 and, if the bins are full, are carried on belt conveyer *36* and *37* (belt width 650 mm, speed 1.6 m/s) into piles *38* and *39*. All piles are isolated from one another by concrete walls, i.e. each grade of gravel is stored separately. Conveyers *23*, *24*, *29*, *30*, *36* and *37* are put in operation automatically by signals of level gauges in the bins *21*, *28* and *35*.

A tunnel is provided under the piles *25*, *26*, *31*, *32*, *38*, and *39*. It contains a belt conveyer *40* (length 350 m, and belt width 1400 mm) and unloading bins with shaking feeders (two for each pile) which can feed the gravel successively from each pile onto the conveyer *40*, which is provided with automatic scales.

Gravel is discharged from the conveyer *40* onto an inclined belt conveyer *41* (length 80 m, and belt width 1400 mm) and carried onto a mobile ('shuttle-type') reversible conveyer *42* (belt width 1400 mm) which is mounted on an unloading bin trestle *43*. The trestle has six bins each of a volume of 200 m^3, according to the number of grades of gravel. The mobile conveyer can move along the trestle and charge gravel into any one of the bins.

The conveyer line *40-41-42* is controlled automatically by signals of top-level gauges in the bins of trestle *43*. When a bin is full, the shaking feeders of the pile being discharged are stopped automatically, and after a certain time

interval, the conveyer *40* and then the conveyers *41* and *42* are stopped. Upon stopping, conveyer *42* moves into a position for charging a next bin and the entire conveyer line is started in the following sequence: the conveyers *42*, *41*, *40* and then the shaking feeders of another pile. It is possible to charge the bins of the bin trestle with only one or two grades of gravel, if required.

Gravel from the bins of the bin trestle is unloaded into railway dump-cars. Thus, gravel can be unloaded from the mill simultaneously into automobiles (from the bins *21*, *28* and *35*) and railway cars (from the bin trestle *43*).

The entire conveyer system is controlled from a central control panel.

18.6. Conveyer System For Charging a Blast Furnace

Modern high-capacity blast furnaces (of a volume of 4000-5000 m³ and more) are charged with materials by means of high-capacity belt conveyers. The conventional cyclic charging by means of skip hoists fails to ensure such a high rate of material supply (1500-2000 m³/h). Attempts to increase the volume of skip buckets and their speed resulted in an excessive increase of the mass and dimensions of buckets and made the skip hoist system inefficient economically. For that reason, a change to belt conveyers has been made in large blast furnaces.

Apart from being highly productive, a conveyer system of blast furnace charging offers a number of advantages of better layout and automatically controlled operation of a blast-furnace shop.

The system (Fig. 18.13*a*) consists of belt conveyers *C1* and *C2* which deliver the components of a blast-furnace charge from a sintering plant and reload them onto a mobile conveyer *C3*, which distributes the loads between bins *1*. The materials from the bins are fed onto screens to screen through fines, after which they are poured into metering funnels each of a volume of 25 m³. The funnels are mounted on scaling devices. The materials are fed from the funnels by means of shaking feeders onto conveyers *C4* and *C5* and then onto an inclined conveyer *C6* which raises the materials to the top of a blast furnace. Emergency repairs of the furnace and conveyer should not take up more than 30 min. Stopping of the conveyer for a longer time is inadmissible in view of the continuous process of metal smelting in the blast furnace.

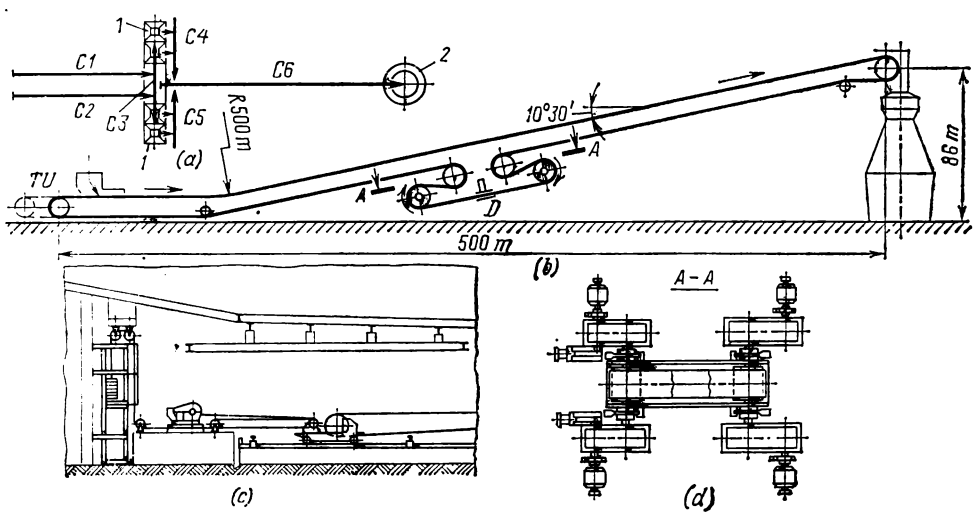


Fig. 18.13. Conveyer for charging blast furnace
(*a*) schematic of conveyer system; (*b*) conveyer *C6*; (c) take-up device; (*d*) drive of conveyer *C6*

Characteristics of Conveyer *C6*

Rated throughput capacity, m^3/h	2000
Belt (rubber-cable, heat resistant)	type PTJI-4000
width, mm	2000
linear mass, kg/m	86
speed, m/s	2.04
Drive (double-pulley type):	
drive pulley diameter, mm	1640
electric motor power kW:	
first pulley	800
second pulley	500
rotational speed of electric motor, rpm	985
Take-up (gravity-winch type):	
tensioning force, kN	300
total travel, m	8
working travel, m	3
weight's mass, t	8.4
Mass, t, of:	
mechanical equipment	376
metal structures	71
belt	93
conveyer	540

Table 18.1. Characteristics of Charge Components Transported by Conveyer *C6*

Component	Density, t/m^3	Lump size, mm
Sinter	1.4-1.6	5-50
Ore pellets	2.1	25
Fluxes (limestone, dolomite, etc.)	1.6-2.8	10-150
Coke	0.43	25-80

The composition of the blast-furnace charge is given in Table 18.1; ore components may have a temperature up to 100°C.

Conveyer *C6* is loaded portionwise with various components of the charge: two portions of coke and two portions of ore and fluxes, each portion having a volume of 25 m^3. The intervals between delivered portions are equal to 17 s. Charge portions are discharged from conveyer *C6* into the cup-and-cone top arrangement of a blast furnace.

The charging of conveyer *C6* and discharging of the cup- and cone arrangement are performed automatically by the control system of the blast furnace shop.

Conveyer *C6* (Fig. 18.13*b*) has a gravity-winch take up *TU* (Fig. 18.13*c*) and a double-pulley drive *D* (Fig. 18.13*d*).

The angle of incline of the conveyer is chosen so as to ensure transportation of spherical ore pellets (of a diameter of 25 mm).

In addition to the main drive, the conveyer has a microdrive for control operations, which moves the belt at a speed of 0.36 m/s. The conveyer is provided with shoe brakes, ratchet-type holdups and a number of sensors which can detect the belt speed, belt deviation from the centre line, condition of belt cables, longitudinal breaks of the belt, and automatic control of the system of circulation lubrication of reducers.

Conveyer *C6* operates continuously without stops all the day round. Planned stops of conveyer *C6* are only possible during the periods of annual repairs of the furnace.

By the process conditions, the conveyer *C6* is not doubled, but has two sets of drive mechanisms (the main and the reserve) which are arranged at both ends of the drive pulleys (see Fig. 18.13*d*). A pulley can be switched over from the main drive to the reserve in a few minutes by switching over the clutches.

The drive compartment contains a spare roll of the belt, which is kept in an inert medium to prevent rubber ageing. These measures along with a high quality of manufacture and assembly of the equipment ensure trouble-free (the availability coefficient $K_a = 0.99$) and uninterrupted continuous operation of the conveyer during a long period (up to 120 days) between repairs.

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Our address is:
USSR, 129820,
Moscow, I-110, GSP,
Pervy Rizhsky Pereulok, 2,
Mir Publishers

Printed in the Union of Soviet Socialist Republics